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# INTEGRATED TECHNOLOGY ROTOR/FLIGHT RESEARCH ROTOR (ITR/FRR) CONCEPT DEFINITION



H. E. Howes
C. A. Tomashofski

Contract DAAK51-81-C-0029
March 1983



NASA

US ARMY AVRADCOM

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Prepared for
U.S. Army Research and Technology
Laboratories (AVRADCOM)
under Contract DAAK51-81-C-0029



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Development Command
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Laboratory
Moffett Field, California 94035

Report No. R-1666
March 1, 1982



# CONCEPT DEFINITION STUDY
# INTEGRATED TECHNOLOGY ROTOR/
# FLIGHT RESEARCH
# ROTOR (ITR/FRR)
# FINAL REPORT

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KAMAN CORPORATION

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## SUMMARY

The concept definition study for the ITR/FRR program was conducted under contract DAAK51-81-C-0029 with the participation of the Aeromechanics Laboratory and the NASA Ames Research Center. Its purpose was to reduce risk in subsequent preliminary design by examining various hub concepts for selection of those with the highest probability of meeting specified goals. Five concepts were studied in a brief preliminary period whereupon two were selected for more thorough development in later phases.

The two concepts selected, the Classic Elastic Pitch Beam (CEPB) and the Plain Elastic Pitch Beam (PEPB), both exhibit superior qualities for the criteria used in the final evaluation. The CEPB is favored over the PEPB and is recommended, primarily because it offers better capability for built-in damping for stability and is judged to have a lower risk in development.

## 1.0 INTRODUCTION

Cost effectiveness and maintainability studies for many years have identified the helicopter rotor head as one of the more complex and expensive aircraft systems. There has long been an ongoing effort, both government and privately sponsored, to develop innovative design techniques and use of advanced materials that would drastically reduce the number of working parts and the weight of rotor head components.

Full-scale flight hardware programs have been quite successful in reducing the number of rotor parts with associated weight savings and development of low cost fabrication techniques. In addition, the cantilevered blades resulting from deletion of hinges have resulted in increased maneuverability. However, systems developed to data have also pointed out new problems to be overcome if the rotor head simplification effort is to succeed. The problems have been primarily associated with stability, performance, increased vibratory hub moments, and structural adequacy. To date, attempts to solve the problems through rework of existing configurations have resulted in only limited success.

The Integrated Technology Rotor/Flight Research Rotor (ITR/FRR) program offers an opportunity to design and develop a totally new rotor system with a solid base of design practices and material usage. Although the ITR/FRR will integrate the latest state-of-the-art in all areas, blade geometry and physical parameters represent a relatively low development risk. The rotor head and associated rotating controls, however, involve a relatively high risk because of the unsolved problems associated with hingeless systems. Because of the degree of risk and the desire for total success of the ITR system, a concept definition phase was initiated ahead of the more extensive preliminary design phase to examine as many rotor head concepts as possible prior to selecting the lowest risk approaches.

There are many varied goals for the ITR/FRR rotor head; however, some of these goals must be singled out for a very critical review because of their major impact on the ultimate acceptance of hingeless rotors. These goals are related to freedom of aeroelastic and mechanical instability, vehicle vibration characteristics, and control response. Other goals, although considered secondary at this time, achieve primary importance to system effectiveness when the primary technical goals are met. They are considered secondary only in that, to some degree, they are a natural fallout of composite hingeless design so that the values assigned to these goals are established to maximize the available benefits. These goals include low drag, weight, and system cost, as well as good fatigue characteristics and reliability and maintainability features. Operational goals include vulnerability characteristics, blade folding, and environmental factors.

Experience in recent years with advanced rotor heads such as the Bearingless Main Rotor (BMR), Starflex, and Triflex has identified the technical problem areas and potential causes. There has been an ongoing effort both analytically and experimentally to examine solutions with varying degrees of success. The objective of the concept definition phase is to implement the results of previous investigative efforts by examining as many rotor head concepts as practical to determine rotor head configurations with the greatest potential for solving the technical problems while preserving and enhancing the attractive life cycle cost features.

The concept definition phase was structured to examine a minimum of five concepts in a brief period and determine two that had the best chance of meeting the varied goals. Those two concepts were then developed to the point of quantifying, approximately, the achievement of goals. A ground rule was established, due to budget and schedule restraints and the nature of the program, that the study would be performed with a combination of conceptual design techniques and engineering judgement with minimum analysis.

To accomplish the many objectives within the necessary restraints imposed, methods and procedures were established to maximize the contribution of specialists in all of the required disciplines. These techniques were successful in establishing a more thorough review of initial concepts than would otherwise be possible. Also, the techniques permitted more analysis of critical areas in the development of the final two concepts, giving more credibility to sizing.

The discussion of the program first describes the assigned Statement of Work tasks as to their intended contribution to the end results. Specific tasks are then discussed in detail.

## 2.0 TASK DEFINITIONS

Following is a list of the specific tasks required by the Statement of Work (SOW):

| | | |
|---|---|---|
| Task I | - | Review Goals And Specifications |
| Task II | - | Selection of Hub Concepts |
| Task III | - | Hub Configuration Development |
| Task IV | - | Determine Physical Properties |
| Task V | - | Evaluation Of Candidate Configurations |
| Task VI | - | FRR Hub Configuration Variations |
| Task VII | - | ITR Compatibility With The RSRA |
| Task VIII | - | Oral Briefing |

Task I - An independent assessment was performed of the various requirements listed as specifications for the ITR/FRR rotor and specifically for the ITR/FRR rotor head. The review was carried out in parallel with the planning and initiation of other tasks.

Task II - This task involved one of the primary efforts and required a brief evaluation of a minimum of five rotor head concepts. The task concluded with the selection of two concepts that had the highest probability of success in meeting all objectives of the ITR/FRR program.

Tasks III, IV, VI, and VII - These tasks could not logically be separated, as they all impacted the development of the two selected concepts. The hub design development required by Task III necessarily had to consider the physical property considerations of Task IV. In addition, the hub variations important to the FRR portion of the program had to be considered as part of the basic design to avoid compromise and limitation to either the ITR or FRR. Task VI involved the rotor shaft/hub interface and the primary control system interface and could not be considered as an independent set of details. The detail requirements of all of the tasks were performed satisfactorily as a unified effort.

Task V - The final evaluation of the selected hub concepts was conducted at the conclusion of all other effort. The evaluation criteria, however, were used throughout the development to guide the design work.

Task VIII - The Final Oral Briefing was held in March 1982 and summarized the findings of the total program.

## 3.0 REVIEW OF GOALS AND SPECIFICATIONS

The contract statement of work contains specifications for both the total ITR/FRR rotor system as well as specific objectives for the rotor head. In most cases, the rotor system specifications refer only indirectly to the rotor head (See Appendix A). Of the various rotor system specifications, those that relate either directly or indirectly to the hub area are:

- Hub flat plate drag area
- Rotor weight
- System cost
- Structural design envelope
- Stability requirements
- Operational requirements such as - blade folding and removal; rain, ice, dust, sand, erosion and lightning protection; limited tree strikes

Combat damage
Maneuverability
Flight test aircraft

No exception was taken to any of the values set as goals for the various factors; however, most of the operational requirements may not be appropriate to an experimental program unless there is serious question as to the need for improvements to the present state-of-the-art.

The contractor has performed the concept definition study on the basis that the ITR and FRR will be flown on the RSRA at a gross weight of 18,400 lbs.

The items designated as Rotor Hub Technical Goals were considered satisfactory for the most part. The value listed for Rotor Hub Moment Stiffness was increased from 100,000 ft-lb/radian to 150,000 ft-lb/radian for a 16,000 lb. aircraft in accordance with discussions with the ATL Technical Monitor.

## 4.0 SELECTION OF CANDIDATE HUB CONCEPTS

### 4.1 Technical Approach

The first major task involved an examination of several rotor head configurations to determine two concepts for further development. The selected concepts would be those that held the most promise of providing satisfactory solutions to fundamental technical concerns in addition to having attractive cost and operational benefits. Considering the technical problems and the severe time limitations, it would have been highly desirable to have a full technical staff devoted to the project. The necessary input from the many engineering disciplines was achieved by adopting a committee-type approach. One designer with good analytical capability with support from the structures department formed the full time support. In addition, senior specialists in dynamics, aerodynamics, stress and design were assigned to participate in weekly meetings to jointly assess problem areas, set priorities, review progress, and determine adequacy of evolving concepts. The full-time design effort and the technical

participation were integrated through strict guidelines set by program management.

### 4.2 Design Methods

Priorities were set to first determine rough sizing for flexible members of each concept, since basic sizing would be the design driver in each case. Preliminary thoughts on interfaces for the blade, rotor shaft, and controls were based solely on engineering judgement with more detailed work left to the development of selected concepts. Rule of thumb stress calculations were applied to beam sizing to determine required geometry for in-plane, out-of-plane, and torsional stiffness as well as buckling strength for static droop. Graphite was used as the beam material for each design with the exception of the compound matrix beam, to maintain a basis for comparison between concepts.

Design criteria including loads, stiffness, and weight distributions were developed through a literature search of recent advanced hingeless rotors (References 1 and 2) and correlation with design data from the advanced rotor study performed for NASA (Reference 3). In addition, analytical background on damping effects through use of elastomer and elastic coupling was updated through a review of References 4-23. Stiffness and bending moment distributions used for preliminary selections are shown in Figures 1 through 6.

### 4.3 Comparison Documentation

To track the many factors used in assessing the relative merits of the concepts, it was necessary to establish a convenient form of documentation. Early attempts were unsuccessful in that grading of a concept for any one factor was easily misunderstood. A status sheet was developed that, with relatively simple written backup, served the purpose. A sample of the status sheet is shown in Figure 7 with a definition of qualification categories following in Table 1. The status sheets were filled out as the conceptual designs developed and served as a focal point of the weekly meetings to structure discussions.

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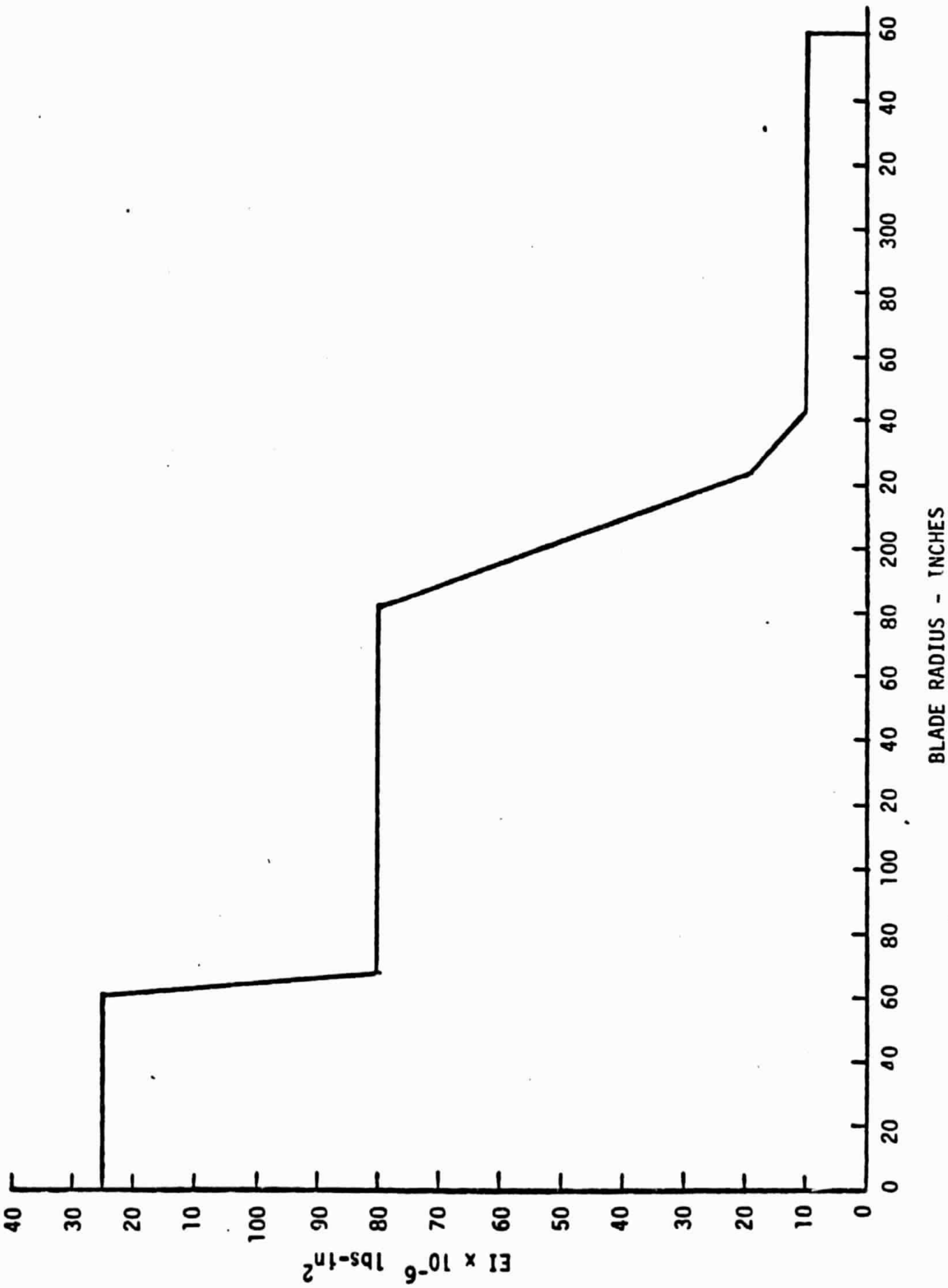


Figure 1. Out-Of-Plane Stiffness.

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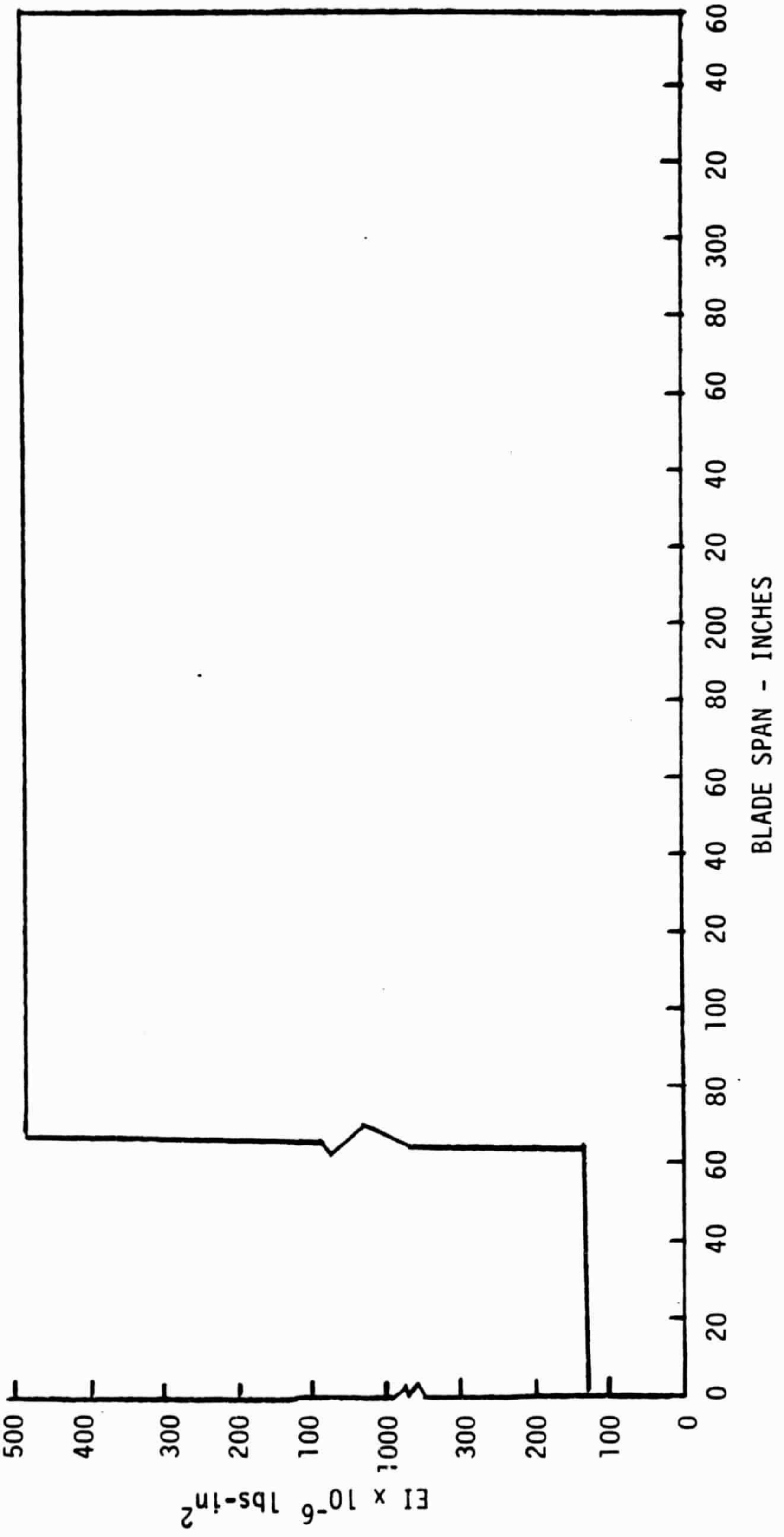


Figure 2. In-Plane Stiffness.

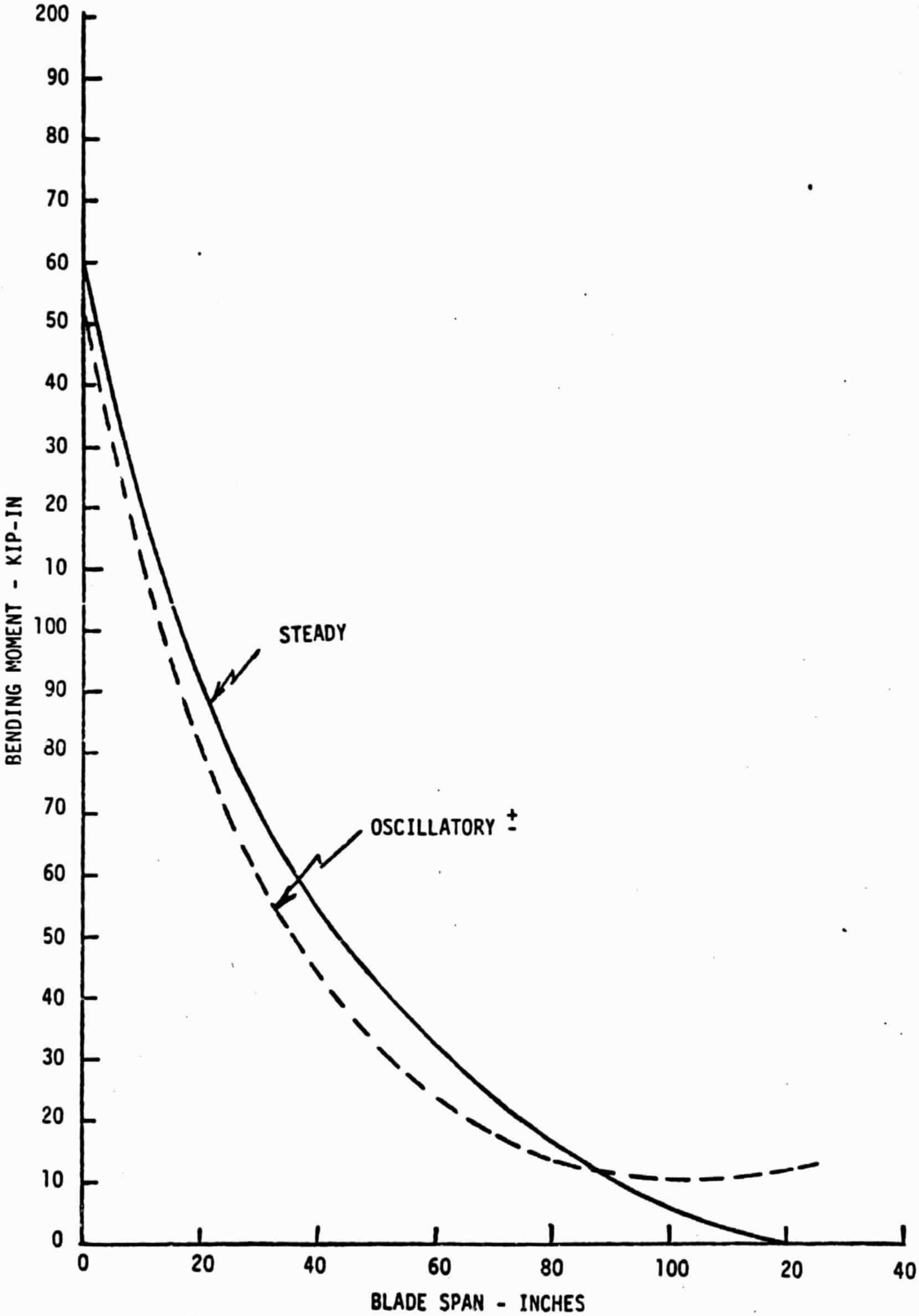


Figure 3. Out-Of-Plane Bending Moment (160 Knots).

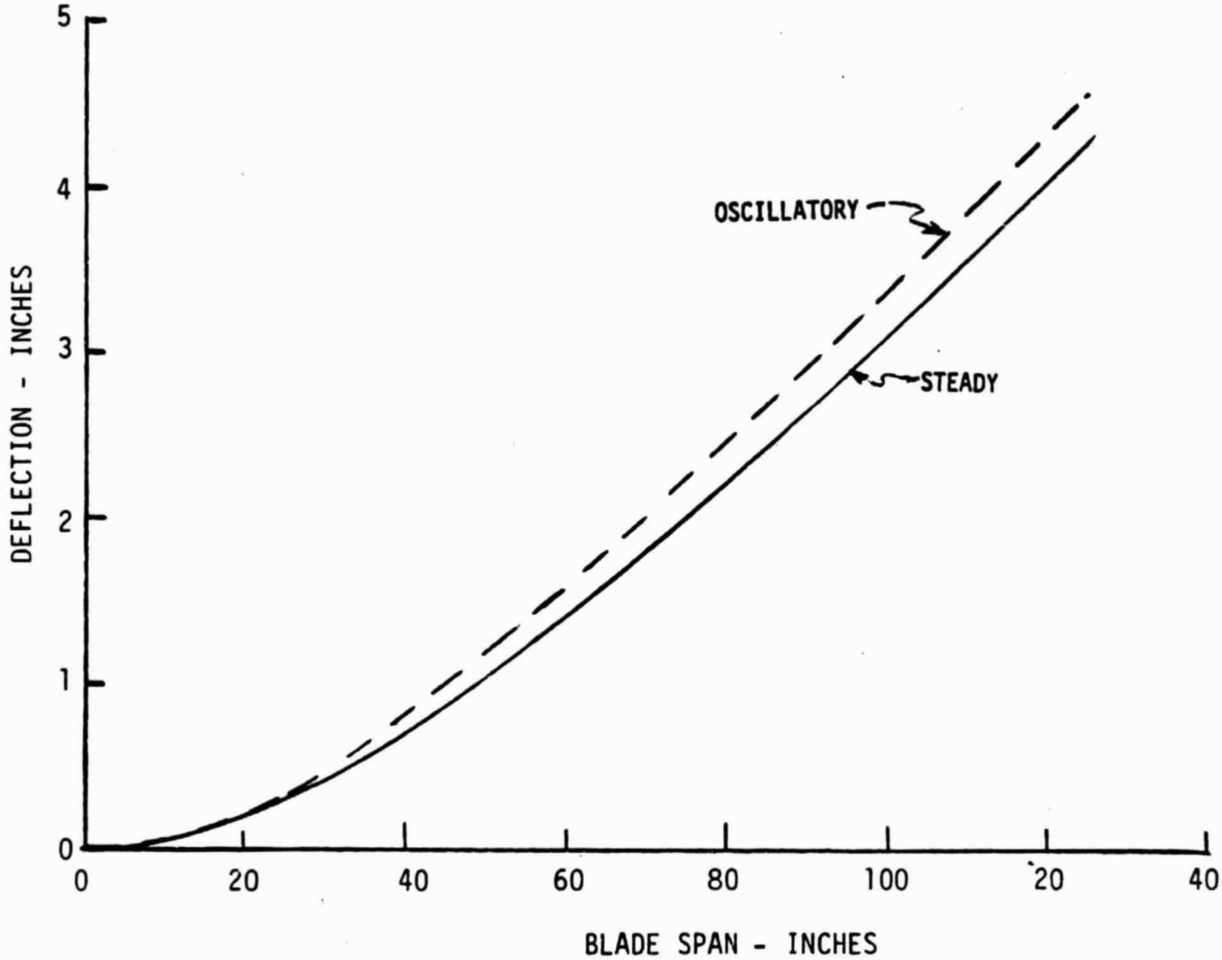


Figure 4. Out-Of-Plane Deflection (160 Knots).

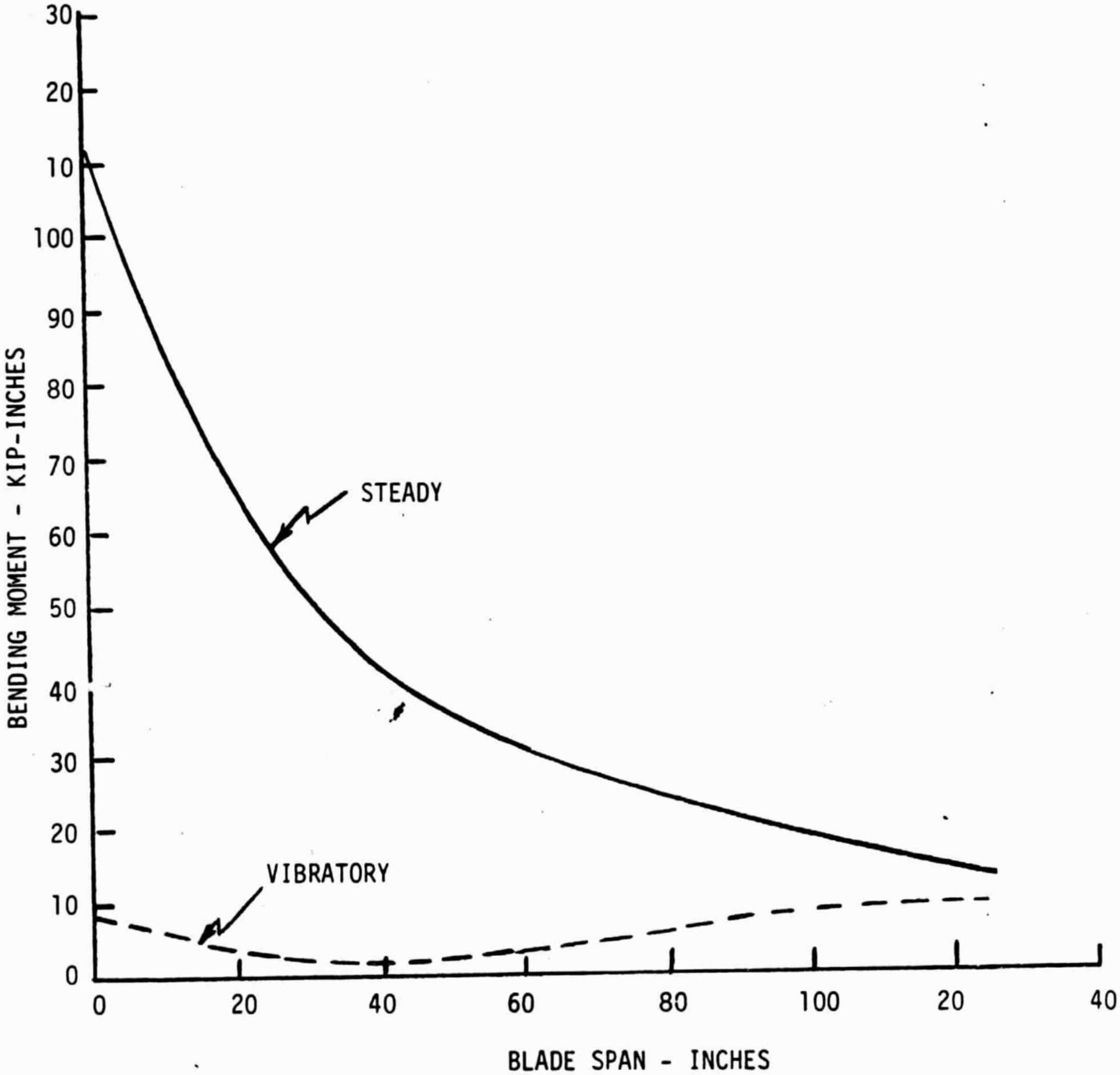


Figure 5. In-Plane Bending Moment (160 Knots).

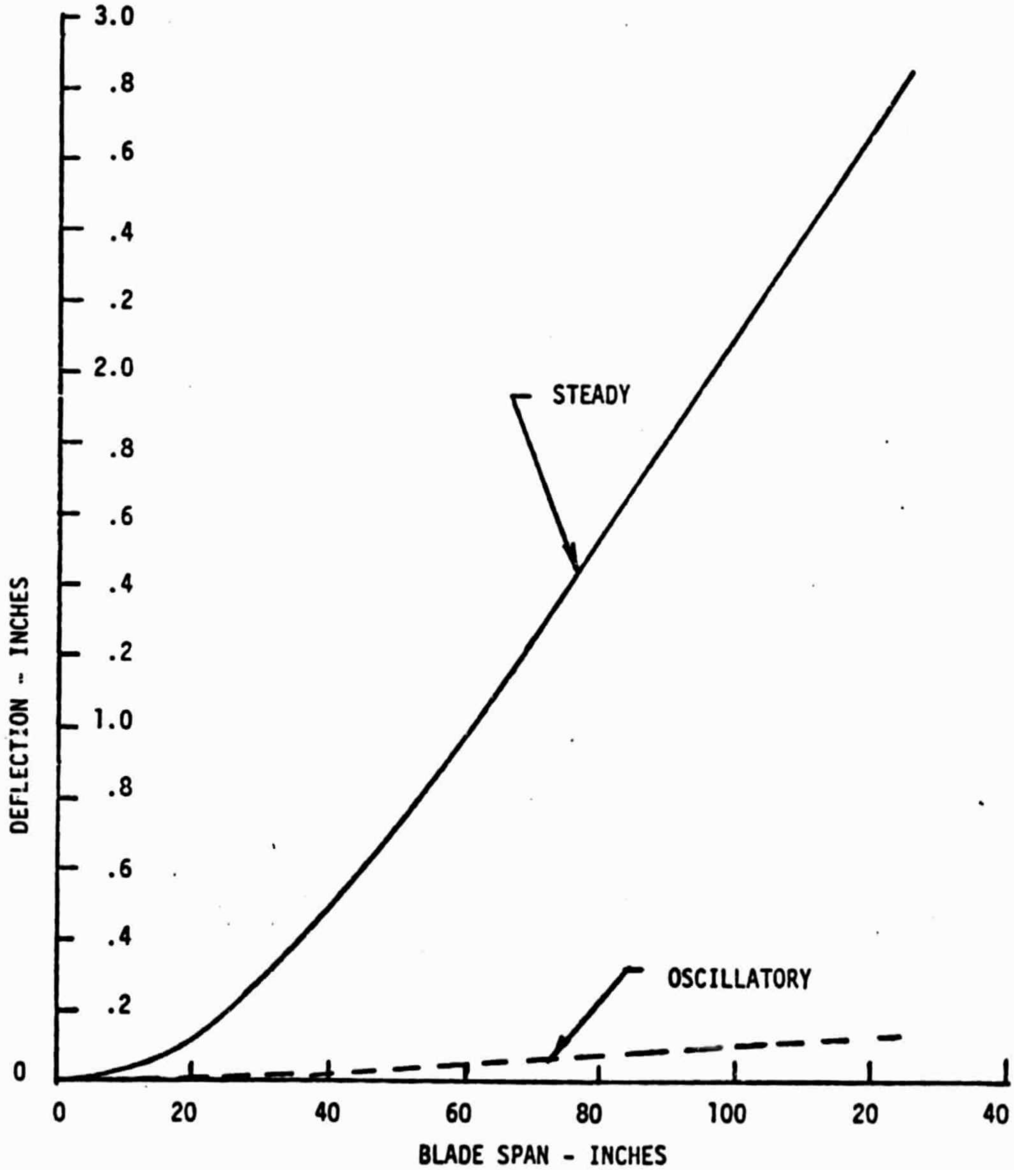


Figure 6. In-Plane Deflection (160 Knots).

| CONCEPT | | | | | | |
|---|---|---|---|---|---|---|
| | DECISION FACTOR | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | | | | |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | | | | | |
| 3 | DAMPING (BUILT-IN)-PARTIAL | | | | | |
| 4 | DAMPING (PITCH-LAG COUPLING) | | | | | |
| 5 | DAMPING (PITCH-FLAP COUPLING) | | | | | |
| 6 | DAMPING (FLAP-LAG COUPLING) | | | | | |
| 7 | LOAD PATHS | | | | | |
| 8 | MECHANICAL LIMITING STOPS | | | | | |
| 9 | DRAG | | | | | |
| 10 | CONTROL LOADS | | | | | |
| 11 | WEIGHT | | | | | |
| 12 | BLADE FOLDING | | | | | |
| 13 | INSPECTABILITY | | | | | |
| 14 | PARTS COUNT | | | | | |
| 15 | COST | | | | | |
| 16 | DEVELOPMENT RISK-FABRICATION | | | | | |
| 17 | DEVELOPMENT RISK-TEST | | | | | |
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Figure 7. Concept Status Sheet.

## TABLE 1. STATUS SHEET FORMAT AND DEFINITIONS

The Status Sheet lists, in the left-hand column, several Decision Factors to be used in judging the merit of a particular concept relative to other concepts under evaluation. Qualification categories are given as headings across the top of the sheet and give a crude indication of merit for a particular Decision Factor. At the far right of the sheet is a column for reference to written explanations of the reasoning for selecting a Qualification Category. Following are definitions of Qualification Categories and Decision Factors:

### QUALIFICATION CATEGORIES

| | | |
|---|---|---|
| 1. | Favorable | A concept has attributes that are obviously very beneficial for the Decision Factor. |
| 2. | Neutral | A concept's attributes are not clearly very beneficial or very poor, but they are not completely unknown. |
| 3. | Unfavorable | A concept has features that are obviously poor or present limits on meeting a goal that are obviously undesirable. |
| 4. | Needs further study | No judgement at all is possible without analysis time that would exceed the preliminary selection schedule (5 concepts to 2 concepts), or that requires analysis beyond the scope of the contract. |

### DECISION FACTORS

| | | |
|---|---|---|
| 1. | Damping (Built-in) - Ample | Means that all damping can be built in without resorting to elastic coupling. Qualification Factors assess the degree of difficulty or compromise to other considerations. |

TABLE 1. (CONTINUED)

| | | |
|---|---|---|
| 2. | Damping (Built-in) - Substantial | Means that almost all damping can be built in, but elastic coupling must be relied on to some extent. Qualification Factors again assess degree of difficulty. |
| 3. | Damping (Built-in) - Partial | Means that 50% or less can be built in. Remainder must be supplied by elastic coupling. Qualification Factors again assess degree of difficulty. |
| 4. | Damping (Pitch-Lag Coupling) | Means that Pitch-Lag coupling is achieved through adjustment of the beam's or beams' orientation. Qualification Factors assess degree of difficulty or compromise. |
| 5. | Damping (Pitch-Flap Coupling) | Same as item 4. |
| 6. | Damping (Flap-Lag Coupling) | Same as item 4. Subject of adjustment of major flexural axis of blade with respect to chord axis is not addressed because it is a blade problem. |
| 7. | Load Paths | Qualification Factors indicate the degree of difficulty anticipated in defining and/or analyzing the structure in the follow-on program. |
| 8. | Mechanical Limiting Stops | Qualification Factors give a measure of the need for droop stops, flapping stops, lag stops, or other mechanical limits. The reference paragraphs will amplify the need. |
| 9. | Drag | Qualification Factors relate only to drag relative to other concepts evaluated. |
| 10. | Control Loads | Qualification Factors relate to load levels relative to RSRA present system capability. |

TABLE 1. (CONTINUED)

| | | |
|---|---|---|
| 11. | Weight | Qualification Factors relate to ITR/FRR SOW specifications. Cannot be treated with any confidence in the first selection of two (2) concepts. |
| 12. | Blade Folding | Qualification Factors relate grossly to anticipated ease of incorporating manual blade folding in a production design without substantial degradation of rotor head benefits. |
| 13. | Inspectability | Qualification Factors relate to amount of disassembly to permit inspection. |
| 14. | Parts Count | Qualification Factors relate only to number of parts relative to other concepts. |
| 15. | Cost | Qualification Factors relate to crude qualitative assessment relative to other concepts based mostly on experience in chosen fabrication techniques. This also relates only to prototype cost and does not attempt to judge production features. |
| 16. | Development Risk - Fabrication | Qualification Factors relate to the degree of uncertainty in the extent of trial cases for process development which will be impacted primarily by new methods to be developed. |
| 17. | Development Risk - Test | Qualification Factors relate to the ability to vary key parameters in test to achieve satisfactory operation. |

## 4.4 Candidate Concepts

4.4.1 Concept 1: Plain Elastic Pitch Beam (PEPB) - The primary flex members of the PEPB are shown in Figure 8. Details of blade, hub, and controls interfaces had not been defined at this point in the study. The flexbeams of the PEPB are in the form of back-to-back "A" frames, filament wound as a continuous structure, to support two blades. The other two blades of the four-blade set are stacked vertically and at ninety degrees. Each beam has plate-like proportions and is uni-directional graphite in an epoxy matrix. The PEPB is referred to as "Plain" because it does not have the structural shell of the "Classic" EPB for transmission of torque for pitch motions.

The PEPB functions as a tension member to react blade centrifugal force, as a flexure to react flap bending, and as a torsion flexure to react blade feathering about the pitch axis. It also reacts in-plane shear and a moment at its outboard end. The torsional stiffness, which must be maintained at a low level for minimum control forces, is comprised of two components, elastic and kinematic. The elastic portion is due to twisting of the beams. Due to the uni-directional fibers, the shear modulus, $G_{23}$, is extremely low compared to the axial modulus, $E_{11}$. Therefore, the elastic torsional stiffness of the PEPB is approximately the stiffness of two edge bars. Additional torsional stiffness can be generated by the kinematic effect, sometimes called "trapeze effect", of turning the axial loads in the two bars. These axial loads originate as reactions to blade centrifugal force, and the effect of turning such large forces (by twisting the outboard end of the PEPB) is to generate significant components normal to the original plane of each bar, such as to form a couple that opposes the twisting displacement. The effect is minimized in the PEPB by causing the two bars to converge at the point of outboard loading so that the turned forces have no arm, neutralizing the kinematic effect.

Pitch is transmitted to the blade by means of a torque tube. The tube could be connected either at the inner apex of the beam as shown in Figure 8 or along the aft wall of the rearmost beam. In either case the tube must be connected through a flexible coupling to permit slope disparity and accept extension changes.

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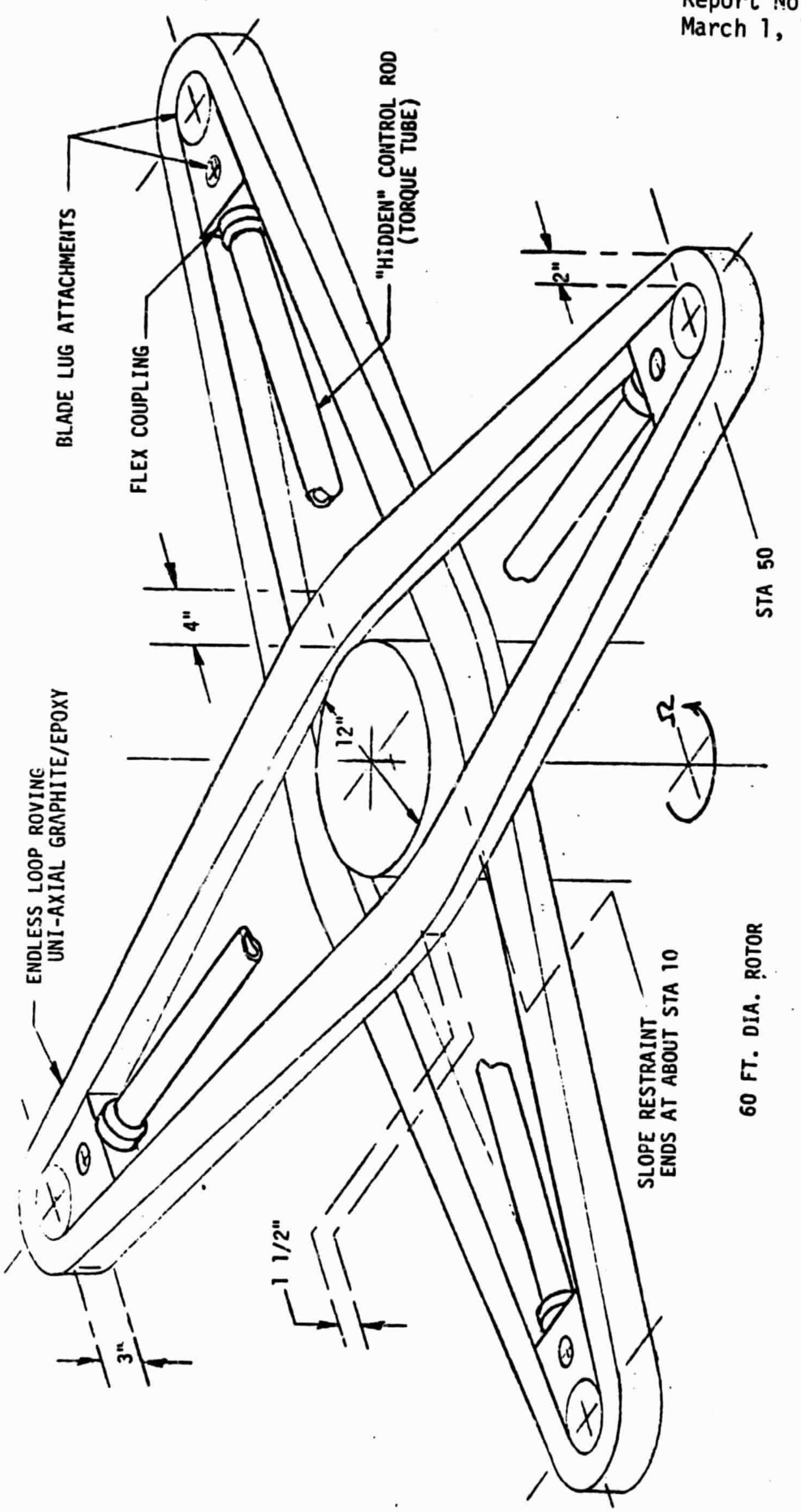


Figure 8. Plain Elastic Pitch Beam, Concept 1

An assessment of the pros and cons of the evaluation factors used is shown in the status sheet of Figure 9.

4.4.2 Concept 2: Plain Elastic Pitch Beam (Elastomer Laminations) - The PEPB with elastomer laminations was the second concept studied. The use of elastomer was looked at primarily for reduction of control forces while maintaining a minimal radial length of the beam to the point of attachment of the blade. Two schemes were reviewed for introducing the elastomer, and both appeared to have very favorable traits for lowering the control forces.

One scheme used elastomer in the form of a cruciform effectively separating each beam into four smaller beams with cross sections approximating squares. The second scheme introduced the elastomer in thin horizontal layers, resulting in a vertical stack of laminations of composite and elastomer. In both cases, the elastomer does not impact primary load paths.

Although both elastomer schemes produce favorable results for lower control forces, the concensus of the engineering committee was that fabrication would very likely be easier with the stacked laminations. In addition, there was a favorable possibility of using the laminations to introduce damping for lead-lag motions.

All other design aspects of the PEPB remained the same. The assessment of all features is shown on the status sheet of Figure 10.

4.4.3 Concept 3: Classic Elastic Pitch Beam (CEPB) - The Classic Elastic Pitch Beam was examined in two configurations, hereafter referred to as Versions A and B. Version A (Figure 11) is the configuration as conceived by Kaman Aerospace several years ago for tail rotors. It is typified by the shell (a structural continuation of the blade) extending from the blade attachment to the rotor shaft attachment and serving to transmit torque for blade pitch.

A single elastomeric joint interconnects the pitch beam and the structural shell at a point close to the rotor shaft and supplies a reaction for shears

| CONCEPT PLAIN ELASTIC PITCH BEAM - SOLID RECTANGULAR BEAM | | | | | | 10/29/81 |
|---|---|---|---|---|---|---|
| DECISION FACTOR | | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | | X | | 1 |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | | X | | | 2 |
| 3 | DAMPING (BUILT-IN)-PARTIAL | X | | | | 3 |
| 4 | DAMPING (PITCH-LAG COUPLING) | X | | | | 4 |
| 5 | DAMPING (PITCH-FLAP COUPLING) | X | | | | 5 |
| 6 | DAMPING (FLAP-LAG COUPLING) | X | | | | 6 |
| 7 | LOAD PATHS | X | | | | 7 |
| 8 | MECHANICAL LIMITING STOPS | X | | | | 8 |
| 9 | DRAG | X | | | | 9 |
| 10 | CONTROL LOADS | | X | | | 10 |
| 11 | WEIGHT | | | | X | 11 |
| 12 | BLADE FOLDING | X | | | | 12 |
| 13 | INSPECTABILITY | X | | | | 13 |
| 14 | PARTS COUNT | X | | | | 14 |
| 15 | COST | | | | X | 15 |
| 16 | DEVELOPMENT RISK-FABRICATION | X | | | | 16 |
| 17 | DEVELOPMENT RISK-TEST | | | | X | 17 |
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Figure 9. Status Sheet - Plain Elastic Pitch Beam, Concept 1

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| CONCEPT PLAIN EPB - ELASTOMER DAMPING, NON STRUCTURAL | | | | | | 1/29/81 |
|---|---|---|---|---|---|---|
| DECISION FACTOR | | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | X | | | |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | X | | | | |
| 3 | DAMPING (BUILT-IN)-PARTIAL | X | | | | |
| 4 | DAMPING (PITCH-LAG COUPLING) | X | | | | |
| 5 | DAMPING (PITCH-FLAP COUPLING) | X | | | | |
| 6 | DAMPING (FLAP-LAG COUPLING) | X | | | | |
| 7 | LOAD PATHS | X | | | | |
| 8 | MECHANICAL LIMITING STOPS | X | | | | |
| 9 | DRAG | X | | | | |
| 10 | CONTROL LOADS | X | | | | |
| 11 | WEIGHT | | | | X | |
| 12 | BLADE FOLDING | X | | | | |
| 13 | INSPECTABILITY | X | | | | |
| 14 | PARTS COUNT | X | | | | |
| 15 | COST | | | | X | |
| 16 | DEVELOPMENT RISK-FABRICATION | X | | | | |
| 17 | DEVELOPMENT RISK-TEST | | | | X | |
| 18 | | | | | | |
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Figure 10. Status Sheet - PEPB - Elastomer Laminations, Concept 2

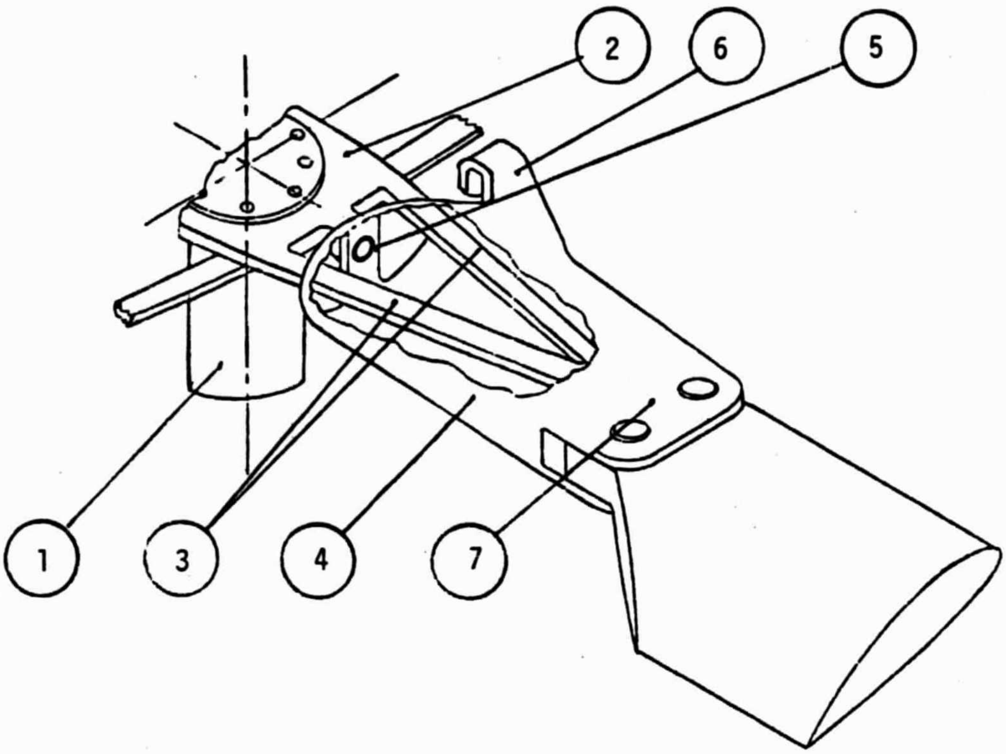
## TABLE 3. PLAIN EPB - LAMINATED WITH ELASTOMER (NON-STRUCTURAL), CONCEPT 2

All Decision Factors are the same as for the PEPB with the solid rectangular cross section, except for Nos. 1, 2, and 10.

1. The ability to introduce a substantial amount of damping by laminating elastomer between the load carrying beam segments is excellent. The magnitude of damping required is unknown at this time resulting in the placement of the check mark in the neutral block. A tradeoff of cost and operational considerations would be necessary to determine whether the laminated PEPB or the solid rectangle with coupling introduced is the better concept.

2. Because of the statements of No. 1, the placement of the check mark as favorable for substantial damping is obvious.

10. Preliminary calculations indicate that control forces reduce drastically with the introduction of elastomer laminations.




| | | | |
|---|---|---|---|
| 1 | ROTOR SHAFT | 5 | ELASTOMERIC PIVOT |
| 2 | HUB CENTERBODY INTEGRAL WITH 3 | 6 | PITCH ARM INTEGRAL WITH 2 |
| 3 | ELASTIC PITCH BEAMS | 7 | BLADE RETENTION INTEGRAL WITH 3 |
| 4 | BENDING/TORSION/FAIRING SHELL | | |

Figure 11. Classic EPB, Concept 3A

normal to the blade span axis. The elastomeric joint is decoupled from the pitch beam and does not react the blade axial loads which are transferred to the pitch beam by two pins connecting the blade to the pitch beam. A pitch horn is connected to the root end of the structural shell to transmit pitch control displacements to the blade assembly. Blade folding is accomplished by removing a single pin at the blade attach point and rotating the blade about the second pin. A clearance cut must be made in the aft wall of the structural shell to permit the blade motion.

The beams of the CEPB function much the same as in the PEPB; however, bending loads are shared with the structural shell. The primary attributes of the CEPB over the PEPB are the potentially smaller beam cross sections, the lower control forces, and an excellent location for lead-lag elastomeric damping at the inboard elastomeric pivot. Also, static droop is not so much of a concern with this concept. A disadvantage is the potentially high drag due to the structural shell.

Because of the high drag, a second configuration, CEPB Version B, was reviewed (Figure 12). This concept is basically the same as Version A except that the structural shell is replaced by an extension of the blade in the form of a clevis, overlapping the outboard end of the beam and joining a torque tube which extends to the elastomeric joint. All of the good features of Version A are retained and flat plate drag area is reduced to approximately that of the PEPB.

The status sheet for CEPB Version B is shown in Figure 13.

4.4.4 Concept 4: Gimballed Hub - The gimballed hub (Figure 14) combines the attractive low torsional stiffness features of the outboard EPB with a bounded flexure attached to the rotor shaft. The composite flexure approximates a gimbal effect and provides the necessary flexibility for flapping. The soft flexure permits tilt of the rotor disc in any direction and permits a low flapping hinge offset without compromising other bending stiffness requirements.

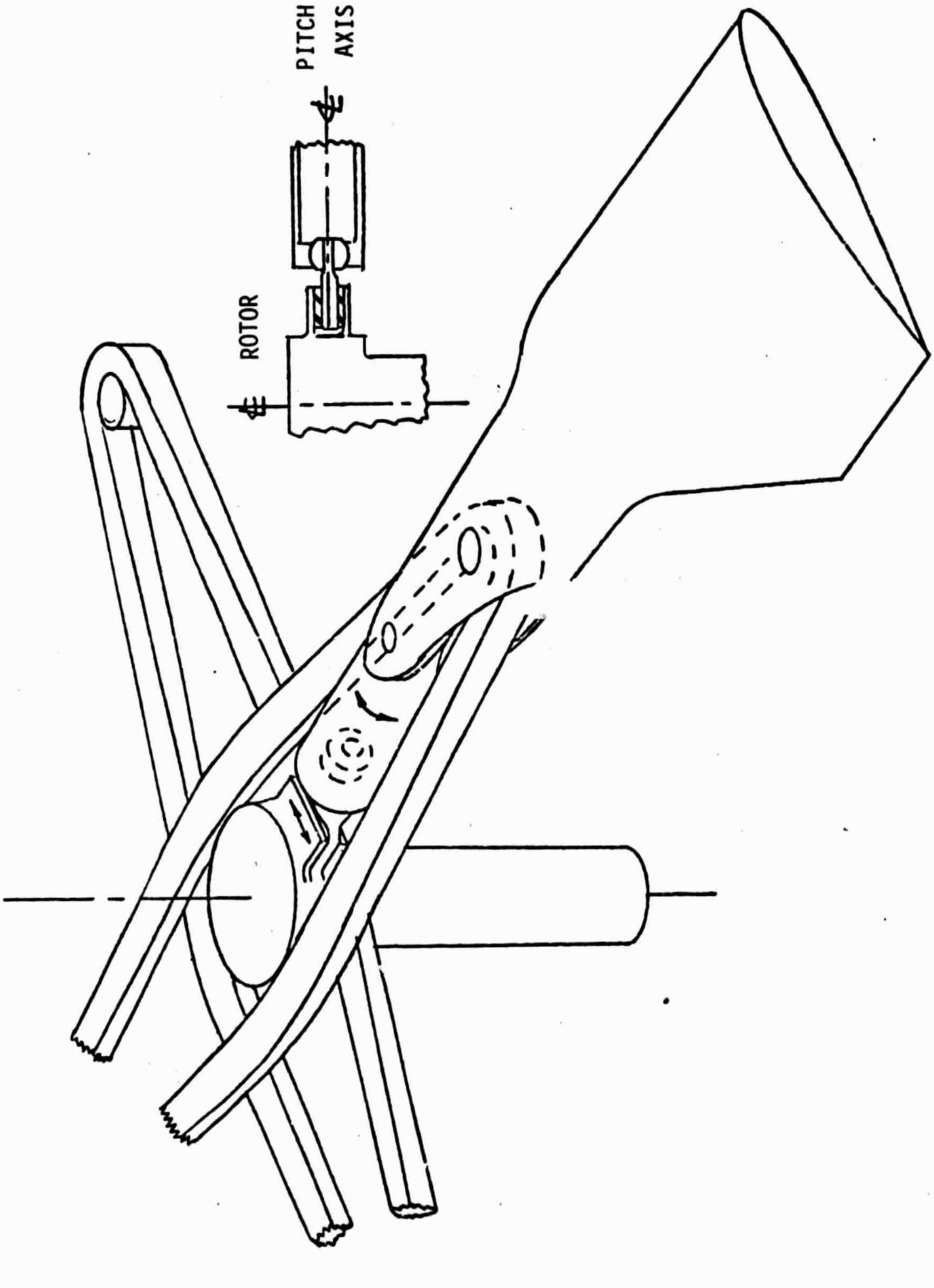



Figure 12. Classic EPB Concept 3B.

| CONCEPT CLASSIC ELASTIC PITCH BEAM - SOLID RECTANGULAR BEAM | | | | | | 10/29/81 |
|---|---|---|---|---|---|---|
| DECISION FACTOR | | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | X | | | |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | X | | | | |
| 3 | DAMPING (BUILT-IN)-PARTIAL | X | | | | |
| 4 | DAMPING (PITCH-LAG COUPLING) | X | | | | |
| 5 | DAMPING (PITCH-FLAP COUPLING) | X | | | | |
| 6 | DAMPING (FLAP-LAG COUPLING) | X | | | | |
| 7 | LOAD PATHS | X | | | | |
| 8 | MECHANICAL LIMITING STOPS | X | | | | |
| 9 | DRAG | X | | | | |
| 10 | CONTROL LOADS | X | | | | |
| 11 | WEIGHT | | | | X | |
| 12 | BLADE FOLDING | X | | | | |
| 13 | INSPECTABILITY | X | | | | |
| 14 | PARTS COUNT | X | | | | |
| 15 | COST | | | | X | |
| 16 | DEVELOPMENT RISK-FABRICATION | X | | | | |
| 17 | DEVELOPMENT RISK-TEST | | | | X | |
| 18 | | | | | | |
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Figure 13. Status Sheet - Classic Elastic Pitch Beam, Concept 3B

## TABLE 4. CLASSIC EPB - SOLID RECTANGULAR BEAM, CONCEPT 3B

1. The elastomeric joint, interconnecting the blade extension and the EPB close to the shaft, provides a location for introduction of significant damping to lead-lag motion. The check mark appears in the neutral block because the magnitude of damping needed has not as yet been determined. It is anticipated that all, or almost all, of the required damping could be introduced at this location.

2. Selection of favorable for substantial damping is self-explanatory based on No. 1.

3. Self-explanatory.

4.
5.
6.
7. Same as Nos. 4, 5, 6, and 7 for the Plain EPB.

8. The Classic EPB has no need for added stops of any kind.

9. Conversion of the structural shell to the blade extension concept results in favorable drag area.

10. Control forces are lighter because the beam cross section is not as large as the Plain EPB.

11. Same as No. 11 for the Plain EPB.

12. Blade folding is more difficult because of the blade extension, however, further development is expected to show this not to be a problem.

13. Inspectability is favorable because the structural shell has been removed in Version B.

TABLE 4. (CONTINUED)

14. Parts count is not as attractive as the Plain EPB, but should still meet the goal of the SOW.

15. Cost. Same as No. 15 for the Plain EPB.

16. Development risk in fabrication is low. All fabrication techniques are well understood and proven.

17. Same as No. 17 for the Plain EPB.

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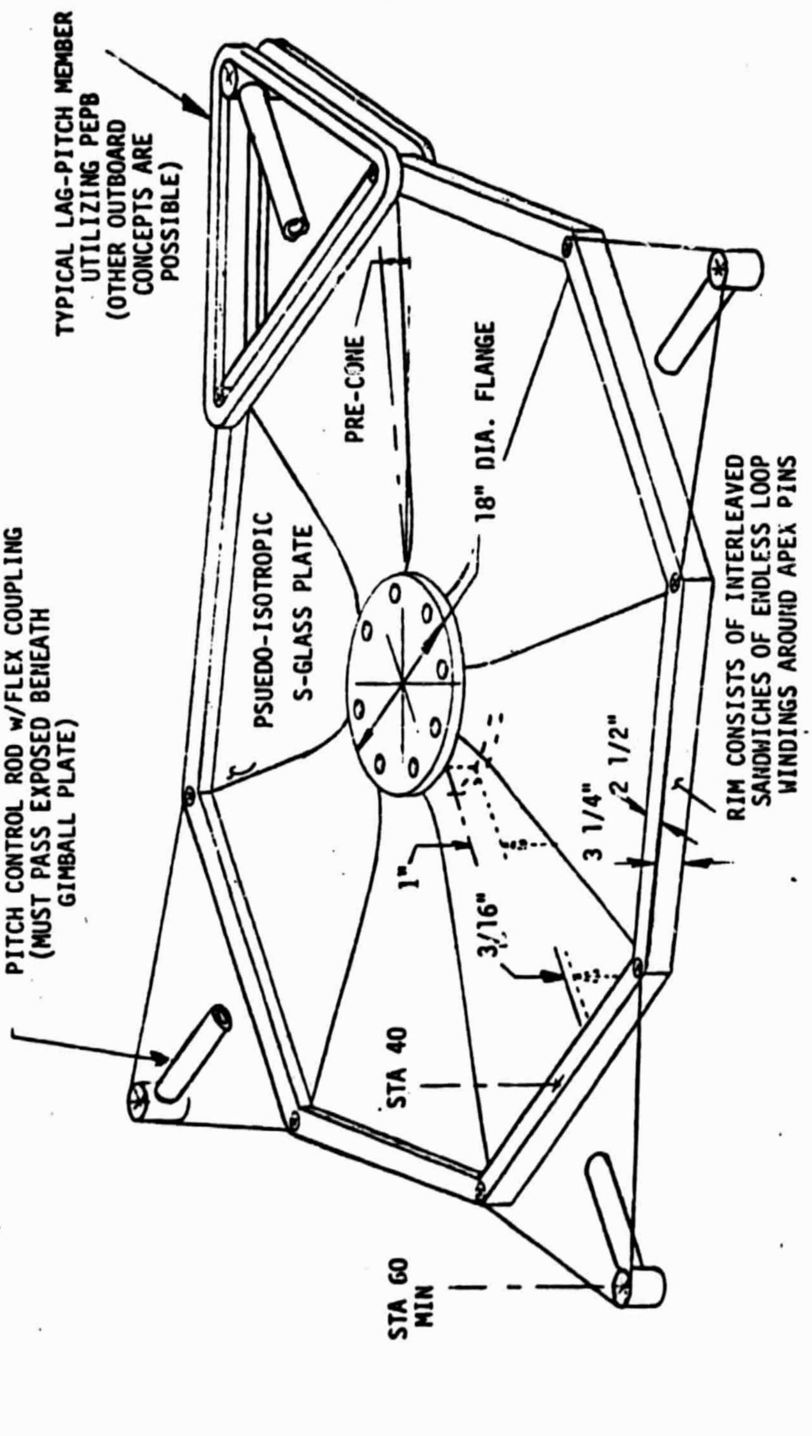


Figure 14. Elastically Gimballed Rotor, Concept 4

The central flexure plate is connected to the rotor shaft flange by a bolt circle, with bolt axis parallel to the shaft axis. Because the plate is thin, the bending stresses induced by the tilt deformations will be moderate. A relatively heavy frame surrounds the central plate and it reacts the bending and axial blade loads using a load path exterior to and redundant with the central plate. These loads are reacted by direct stresses in the frame and not by bending stresses. Therefore, the frame is proportioned for high stiffness and carries the major share of these loads so that major tension stresses would not be generated in the central disc.

The EPB outboard of the gimbal interconnects the blade and gimbal frame. The EPB, in this case, would be proportioned primarily for low torsional stiffness and the appropriate stiffness levels in bending in its two planes without the emphasis of a soft flexure for flapping.

The gimballed hub is a relatively complex structure to examine, even based on judgement, in the time permitted. Emphasis was placed on rough sizing the gimbal and using the combined engineering judgement of all specialists in assessing the remaining component parts and functions.

The gimballed hub has an attractive feature of providing a relatively low flapping hinge offset while separating the various other required functions for ease of tailoring characteristics. Areas that are open to question and not easily assessed without further work include potentially higher drag, cost, and parts count. The ability to introduce elastomeric damping or elastic coupling effects could be hampered compared to other basic EPB concepts.

The assessment based on the limited work performed is shown on the status sheet in Figure 15.

4.4.5 <u>Concept 5: Compound Matrix Beam</u> - This concept (Figure 16) attempts to combine all required functions in a single beam. As in the EPB concepts, the beams are stacked vertically. Meeting the many varied requirements demands a highly specialized composite material. Not only is the composite highly

| CONCEPT GIMBALLED HUB - EPB ASSUMED OUTBOARD | | | | | | 10/30/81 |
|---|---|---|---|---|---|---|
| | DECISION FACTOR | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | | X | | |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | | X | | | |
| 3 | DAMPING (BUILT-IN)-PARTIAL | X | | | | |
| 4 | DAMPING (PITCH-LAG COUPLING) | | | | X | |
| 5 | DAMPING (PITCH-FLAP COUPLING) | | | | X | |
| 6 | DAMPING (FLAP-LAG COUPLING) | | | | X | |
| 7 | LOAD PATHS | | X | | | |
| 8 | MECHANICAL LIMITING STOPS | X | | | | |
| 9 | DRAG | | X | | | |
| 10 | CONTROL LOADS | | | | X | |
| 11 | WEIGHT | | | | X | |
| 12 | BLADE FOLDING | | X | | | |
| 13 | INSPECTABILITY | X | | | | |
| 14 | PARTS COUNT | | X | | | |
| 15 | COST | | | | X | |
| 16 | DEVELOPMENT RISK-FABRICATION | | X | | | |
| 17 | DEVELOPMENT RISK-TEST | | | | X | |
| 18 | | | | | | |
| 19 | | | | | | |
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Figure 15. Status Sheet - Gimballed Hub, Concept 4

TABLE 5. GIMBALLED HUB - EPB ASSUMED OUTBOARD, CONCEPT 4

1. Built-in damping, through addition of elastomer, only appears feasible in the EPB outboard of the gimbal. The radial span of the EPB section, when compared to the Plain EPB concept appears much shorter, limiting the beam length available for effective damping. Therefore, the check appears in the unfavorable box indicating little chance of achieving all necessary damping.

2. There is some uncertainty that substantial damping could be achieved in this concept.

3. Some percentage of damping through elastomer has a high probability, hence, the favorable check.

4.
5.
6. It is uncertain how much elastic coupling of any kind could be provided without further study.

7. Although load paths can be identified with this concept, the structural analysis appears more complex than the various versions of the EPB concept.

8. No limit stops should be necessary in this concept.

9. Drag appears worse in this concept that the Plain EPB, or the Classic EPB, therefore, the check is temporarily in neutral.

10. Control loads have not been estimated and additional analysis would be required.

11. Weight cannot be addressed for any concept with any confidence during the preliminary selection process.

TABLE 5. (CONTINUED)

12. Blade folding could be more difficult than the Plain EPB because of the space restriction due to the gimbal.

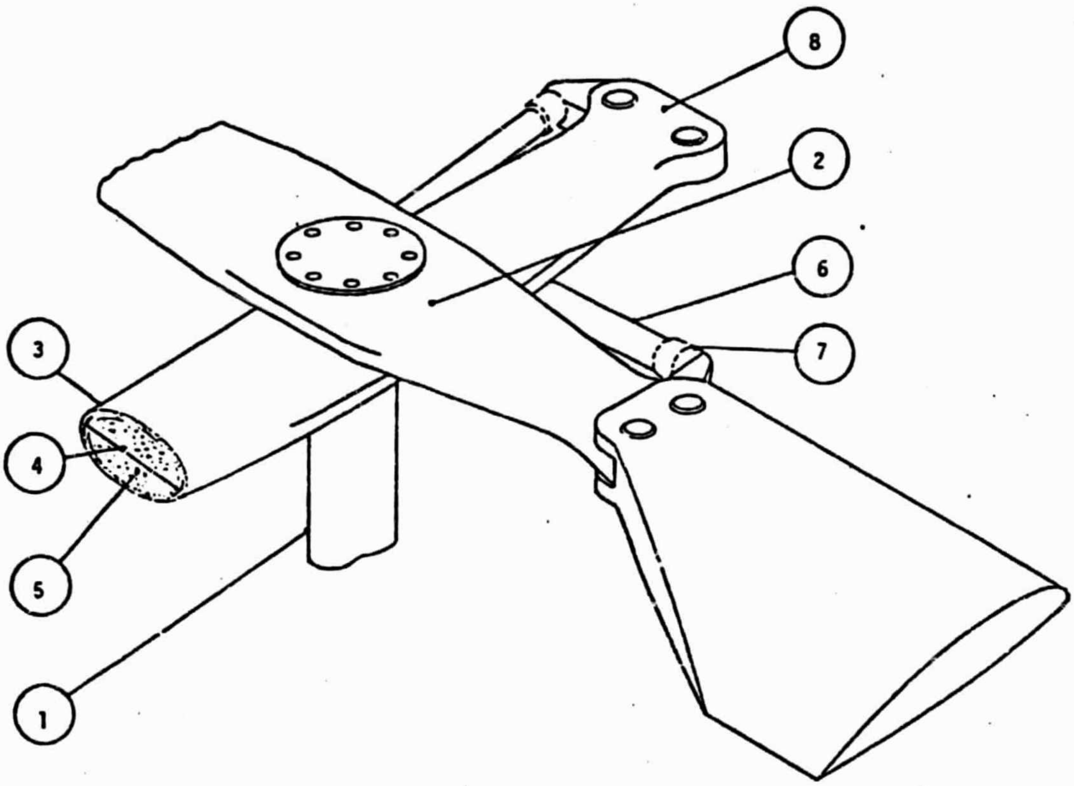
13. Inspectability is good with this concept. All critical components are exposed for easy viewing.

14. Parts count, while not as low as the Plain EPB, will probably meet the specification goal.

15. Cost cannot be assessed at this time, except that it is almost certain to be higher than the Plain EPB.

16. Development risk in fabrication is relatively low. The gimbal plate is a more sophisticated structure than the EPB, but similar plate-like composite structures have been designed, fabricated and tested at Kaman in the past.

17. Development risk in test cannot be addressed at this time.




1 ROTOR SHAFT

2 HUB CENTERBODY FLEXURE REGION (EPOXY MATRIX)

3 TORSION/BENDING ARM-ELASTOMERIC & EPOXY MATRICES

4 EPOXY COMPOSITE SHEAR WEB AND FLANGES

5 PLASTIC FOAM STABILIZING CORE

6 PITCH CONTROL TORQUE TUBE

7 KAFLEX (TM) ELASTIC COUPLING

8 INTEGRAL BLADE RETENTION

Figure 16. Compound Matrix Pitch Beam, Concept 5

specialized but its characterization must vary in different spanwise sections to supply either rigidity or softness, dependent on function. In addition, any provision for structural damping must be inherent in the composite itself; otherwise, elastic coupling must be relied on solely to supply the necessary damping for lead-lag motion.

As stated above, the concept involves separation of various zones along the length of the beam for various functions, tailoring the sections by geometry and inherent properties of structural material. The section immediately outboard of the central hub approximates a flat plate in section to accommodate flapping flexibility while maintaining high edgewise stiffness, although it may be difficult to achieve the low hinge offset desired. The composite in this section is S-glass fiber in an epoxy matrix. Outboard, the section transitions to an elliptical shape and the S-glass fibers transition to bundles or rods of S-glass and epoxy. Torsional stiffness is lowered in this area by imbedding the S-glass rods in an elastomeric matrix. At the blade attachment area, the section transitions again to a flat plate of S-glass and epoxy. A glass-epoxy shear web will very likely be carried through the center of the beam throughout the span. A material such as plastic foam might also be used as a stabilizing core, although this has not been determined to be necessary.

Pitch control would be achieved through a conventional torque tube approach with a flexible coupling at the blade attachment point for bending and extension displacements.

Some damping for lead-lag motions through tailoring of the elastomer matrix will be achieved, but a significant contribution from this source cannot be relied on, and it must be recognized that the major contribution will very likely come from elastic coupling effects. There are more limitations in this concept to the means for introducing such coupling than in the EPB configurations.

Another important factor is the development risk in fabrication considering the complex composite structure. A significant number of specimen tests will be

required in addition to analysis to predict characteristics, and there will still be risk in achieving predicted characteristics in full-scale specimens without a number of trials.

Production cost may or may not be low, depending on the amount of handwork to set up the complex structure. It is difficult to project a fully automatic process that would accomplish the many necessary steps. Quality control in a production article can also be expected to be a problem.

All of the evaluation factors are addressed in the status sheet of Figure 17.

### 4.5 Hub Selections for Further Development

All of the rotor heads examined have features that, with adequate development, could make them good candidates for cost effective systems that would perform satisfactorily. The two concepts selected, however, clearly have features that can lead to satisfactory achievement of all primary goals and very likely the secondary ones as well. Concept 2, the PEPB with elastomer laminations, and Concept 3B, the CEPB Version B, were selected. The only real doubt for either of these concepts is whether the PEPB with more accurate sizing can handle the requirements for static droop and moment at the apex without growing in size and degrading drag and weight concerns.

Concept 1, the PEPB without laminations must grow in length to reduce control forces and thus, degrades many of the goals. Control force has been shown to be a major design driver. If the beam length is maintained (the high control forces accepted), then all components and attachments of the rotating and stationary hydraulic control system must grow, again degrading goals. Undesirable trade-offs of this nature can result in decisions that hingeless rotor systems are not cost effective, and thus Concept 1 was discarded.

Concept 4, the gimballed hub, might be a very good answer to the questionable areas of hingeless systems but requires considerably more effort than permitted under the present contract to prove it with any confidence.

| CONCEPT COMPOUND MATRIX BEAM - PRELIMINARY | | | | | | 10/29/81 |
|---|---|---|---|---|---|---|
| | DECISION FACTOR | FAVORABLE | NEUTRAL | UNFAVORABLE | NEED FURTHER STUDY | REFERENCE PARAGRAPH |
| 1 | DAMPING (BUILT-IN)-AMPLE | | | X | | |
| 2 | DAMPING (BUILT-IN)-SUBSTANTIAL | | X | | | |
| 3 | DAMPING (BUILT-IN)-PARTIAL | X | | | | |
| 4 | DAMPING (PITCH-LAG COUPLING) | | | | X | |
| 5 | DAMPING (PITCH-FLAP COUPLING) | | | | X | |
| 6 | DAMPING (FLAP-LAG COUPLING) | | | | X | |
| 7 | LOAD PATHS | X | | | | |
| 8 | MECHANICAL LIMITING STOPS | X | | | | |
| 9 | DRAG | | X | | | |
| 10 | CONTROL LOADS | X | | | | |
| 11 | WEIGHT | | | | | |
| 12 | BLADE FOLDING | X | | | | |
| 13 | INSPECTABILITY | | X | | | |
| 14 | PARTS COUNT | X | | | | |
| 15 | COST | | | | X | |
| 16 | DEVELOPMENT RISK-FABRICATION | | | | X | |
| 17 | DEVELOPMENT RISK-TEST | | | | X | |
| 18 | | | | | | |
| 19 | | | | | | |
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Figure 17. Status Sheet-Compound Matrix Beam, Concept 5

## TABLE 6. COMPOUND MATRIX BEAM - PRELIMINARY, CONCEPT 5

1. A limited amount of built-in damping should be inherent in this concept because of the elastomer matrix. Because the magnitude of damping needed is unknown, it is impossible to say whether or not a significant amount of damping could be supplied.

2. From the statement in No. 1, it is possible that substantial built-in damping will be present.

3. Reference statement No. 1.

4.
5.
6. It is difficult to say at this time how much elastic coupling can be provided without compromising some areas of the operating envelope.

7. Load paths appear to be easily definable although this may not be true due to the complex structure of the beam.

8. No limiting stops should be necessary in this concept.

9. The check mark is in neutral for drag because the concept probably has higher drag than the Plain EPB but may not be worse than the other concepts.

10. Control loads are probably higher than the Plain EPB with elastomer laminations but without some analysis it is difficult to assess. They can be expected to be reasonably low.

11. Weight cannot be assessed for any concept at this time.

12. Blade folding characteristics should be as good as the Plain EPB.

TABLE 6. (CONTINUED)

13. Inspectability of exterior surfaces should be good but integrity of the complex material matrix can only be checked by NDI techniques such as X-ray.

14. Parts count will be approximately the same as in the Plain EPB. Both the Plain EPB and the Compound Matrix Beam should have the lowest parts count of all concepts reviewed.

15. Cost cannot be assessed in the preliminary selection period.

16. Development risk in fabrication requires further study but is very likely to be higher than the Plain EPB or the Classic EPB because of the complex structure. More trials in process development will probably be necessary to achieve desired properties.

17. Development risk in test cannot be assessed at this time.

Concept 5, the compound matrix beam, is in the same category as the gimballed hub except that it has the additional disadvantage that there could be considerably more development risk, both in fabrication and in test. It is very likely that a reasonably good rotor head concept could evolve with the compound matrix beam. Past history has indicated, however, that a workable solution with such a complex structure only results from an expensive trial and error flight test approach.

Considering the factors cited, Concept 2 and Concept 3B were logical choices for development. With a well planned development program at least one of these concepts should produce an advanced rotor head system that answers present needs. There should be relatively low risk both in fabrication and test. The various functions can be reasonably well predicted and low cost variations for flight test can be planned for any doubtful areas.

## 5.0 ROTOR HEAD DEVELOPMENT - SELECTED CONCEPTS

The preliminary selection process concentrated primarily on sizing of flexures for bending and torsional requirements as this was felt to be the design driver. The development effort split into two tasks to gain maximum benefit from the time available. One of these tasks involved conceptual design, concentrating on all of the interface areas, i.e., blade-to-beam, beam-to-rotor shaft, and torque tube to rotating swashplate. Beam size was kept the same as in the preliminary selection period, pending more accurate analysis. Emphasis was placed on such factors as drag, weight, parts count, maintainability and cost.

The second task involved working with a simplified finite element analysis to more accurately size the beams and torque tubes. Also, various methods of altering the beam geometry were briefly analyzed to determine structural impact and cross coupling effects from introducing the different forms of elastic coupling. The use of elastomer laminations was also studied for effect on control force and hub stiffness, and a brief review of the use of graphite, Kevlar, and S-glass was accomplished.

As the analysis and the conceptual design effort progressed, the various features of the CEPB and the PEPB merged until, for all practical purposes, they became one concept. It was found that the best interfaces for blade attachement, rotor shaft attachment, and pitch horn control were the same for both concepts. It was necessary to introduce elastomer laminations in the beams of each concept to lower control forces for minimum beam length and to tailor the placement of the equivalent hinge offset. The major difference between the concepts finally was the increased size of the unsupported torque tube of the PEPB; and with a potential start-up problem it seemed prudent to support the tube in a sliding pivot bearing, as in the CEPB, and take advantage of the location for elastomeric damping. Thus correction of the PEPB weaknesses, in effect, changes it into a CEPB. All of these factors are discussed in more detail in subsequent sections describing the analytical and design results.

The elastic pitch beam in its final configuration (after development) is shown in Figures 18 through 20. Within the limited design and analytical effort, it appears that all design goals have been met or exceeded with the exception of cost. The cost target and baseline, however, were somewhat arbitrary.

The configuration shown in Figures 18 through 20, when considered as a CEPB, should, perhaps, be redesignated as Concept 3C. It is unique from either Concept 3A or 3B in that, like Concept 2, the beams have been leaved with elastomer laminations. When considered as a PEPB, it may still be correctly designated as Concept 2.

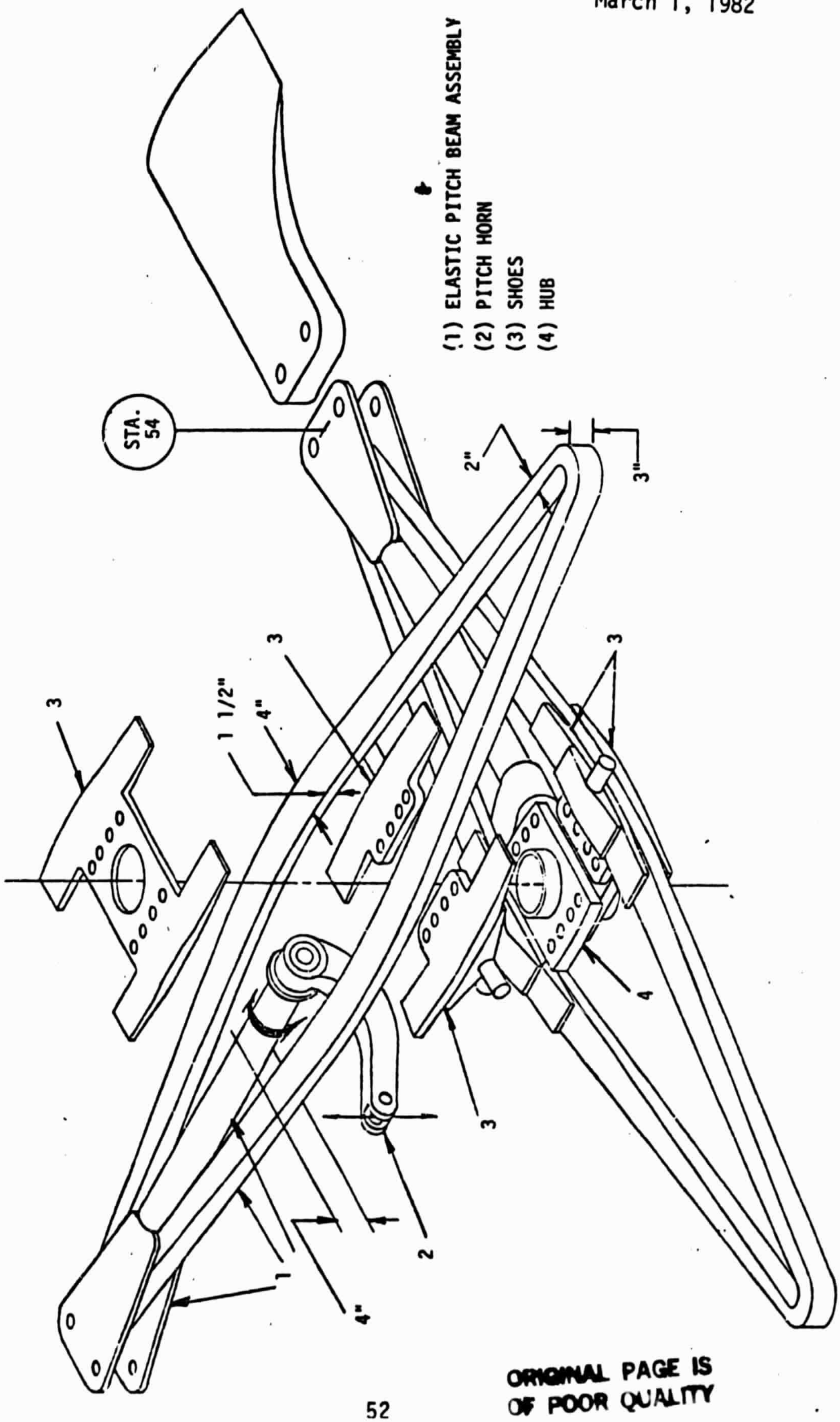


Figure 18. Elastic Pitch Beam Rotor Head - Exploded View

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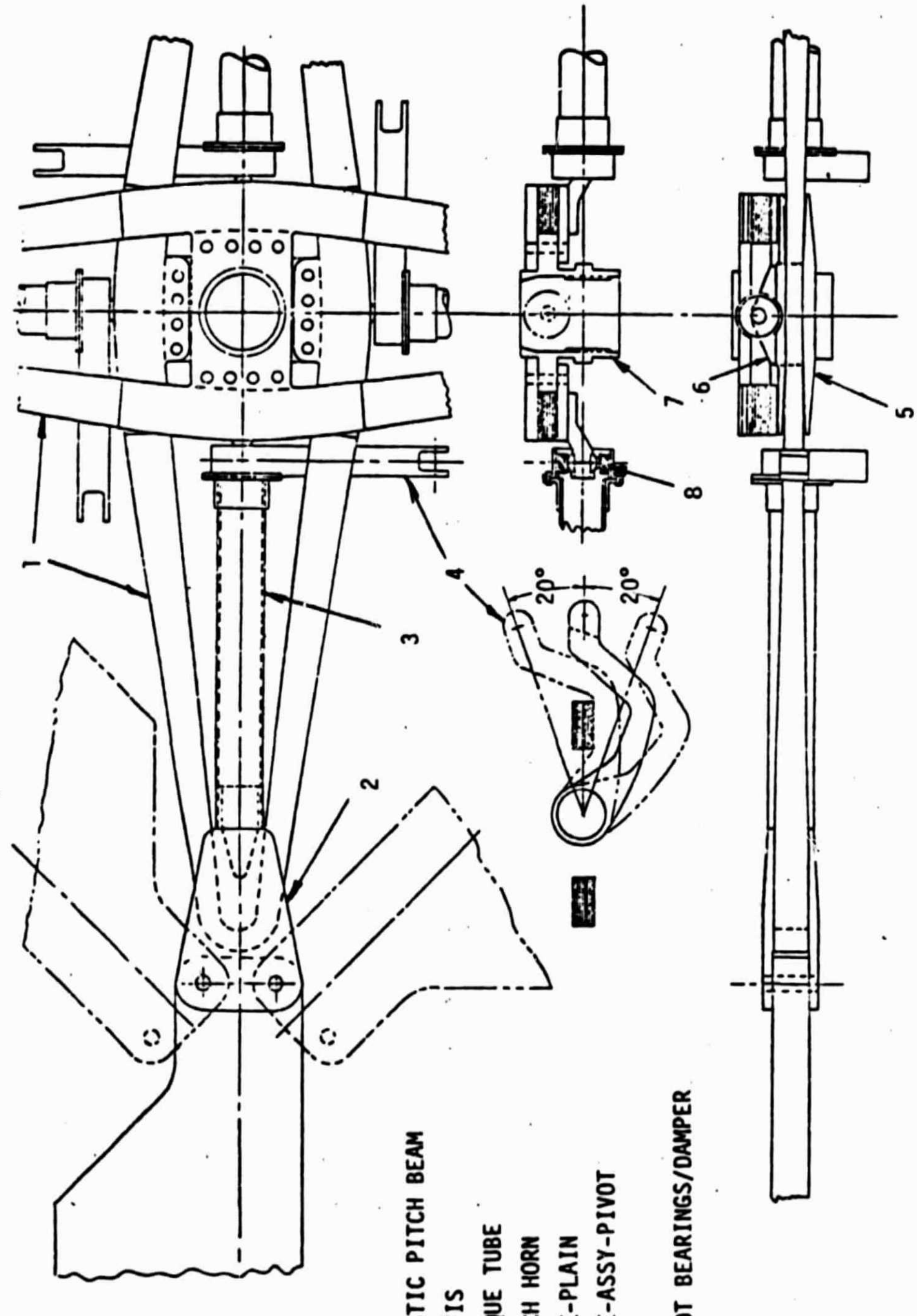


(1) ELASTIC PITCH BEAM
(2) CLEVIS
(3) TORQUE TUBE
(4) PITCH HORN
(5) SHOE-PLAIN
(6) SHOE-ASSY-PIVOT
(7) HUB
(8) PIVOT BEARINGS/DAMPER

Figure 19. Elastic Pitch Beam Rotor Head - Plan View & Sections

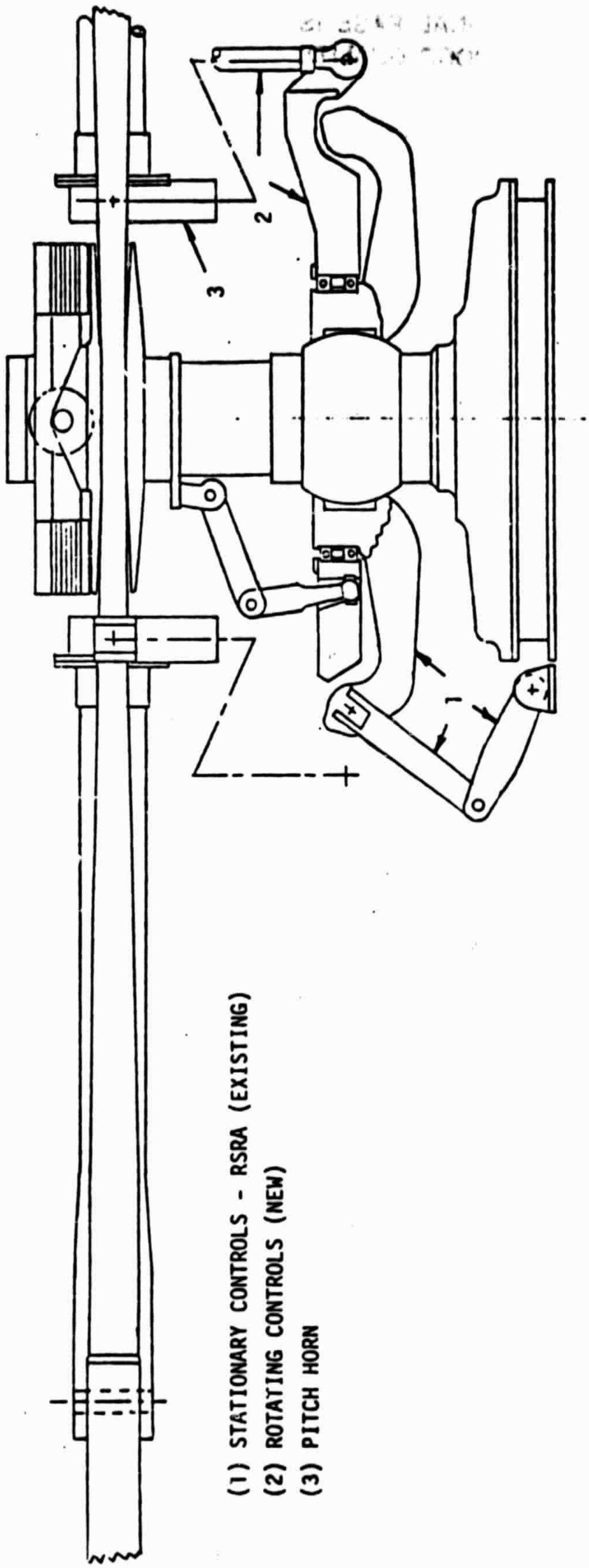


Figure 20. Elastic Pitch Beam Rotor Head - Controls.

## 5.1 Simplified Structural Analysis And Design Considerations

5.1.1 Scope And Intention Of Analysis. The first priority was to define the critical criteria which drive the component sizing. Preliminary investigations early in the program demonstrated that static droop and "hard startup" stand out as being critical. Both of these conditions occur when centrifugal force is not present and thus no relief of compressive bending stress can be invoked. In each case, these two criteria were monitored as the basic sizing devices.

Definition of a baseline material was the next obvious step. Uniaxial fiber advanced composites are implicit in the elastic pitch beam concept; therefore, the choice was among graphite, fiberglass, and Kevlar. Evaluation was done by setting down desirable quantities - in the numerator if large values were desirable - in the denominator if small values were desirable - into a qualitative merit factor for each material.

After rough sizing the baseline elastic pitch beam for each configuration - classic and plain - a fast, simple-to-use mathematical technique was chosen to obtain deflected shapes and stresses for a large number of cases. The accuracy of the results is expected to be to first order. The technique was then exercised to corroborate the rough sizing results and to assess the effects of applied control loads, structural coupling, and the various materials.

Finally, certain recommendations were made based on the results.

5.1.2 Some Precursory Requirements And Considerations. The high centrifugal tension places an absolute lower limit on the total cross-sectional area of the retention beams. For this study, it was set at

$$\underline{3.05\ \text{in}^2} = \frac{(61\ \text{kips})(1.5\ \text{ult})}{(30\ \text{ksi})} \qquad \text{i.e.,} \qquad A = \frac{(\text{CF})(\text{Ult. Factor})}{(\text{Tensile Allowable})}$$

In-plane excursions of the blade tip must be monitored and damping of that motion must be provided for good ground stability characteristics. Out-of-plane excursions must be minimized to allow blade/fuselage clearance under multiple-g static droop without the need for extra droop stop hardware. This latter consideration, however, should be made secondary to the requirements of low flap hinge offset and control forces within current RSRA capability. It is subsequently shown that with the elastic pitch beam concept, these requirements are not incompatible and droop stops are not necessary. It is also desirable to "build in" aerodynamic damping by ensuring that structural coupling enhances stability.

Finally, simplicity of the conceptual design should be a background concern at all times, reflecting anticipation of the normal growth in complexity as a workable final design is realized. While this philosophy is inherent in most design projects, it is specifically noted here since one major goal of the ITR is to simplify maintenance and inspection and to increase reliability of the rotor.

5.1.3 Baseline Definition. In understanding the rationale behind the baseline definition, it is helpful to note the advantages of the A-frame elastic pitch beam. Unlike some past experimental bearingless rotors, the EPB allows for a high degree of manufacturing automation. Endless loop winding of uniaxial fibers across the center from blade to blade eliminates the need for centrifugal retention hardware at the hub. Further, the composite is loaded mainly along its strongest axis, and no bolts need breach the filaments - frequently the weak link in composite designs. The presence of a virtual apex slightly outboard of the blade retention clevis minimizes the trapeze effect. This is the torsional stiffening of a member with a nonconnected cross-section when an axial force is applied. Finally, the open planform of the A-frame allows the controls torque tube to be well hidden aerodynamically.

The term "cross-section tailoring" is used here to mean variation of the aspect ratio of the individual beam elements along their span. A rectangle was chosen as the simplest, most logical shape for this type of member and is quite natural

for the winding operation intended. Proper tailoring of these rectangles can effectively separate the flapping and lagging hinges. By thinning the vertical dimension near the root, the majority of flap bending will take place inboard for a low effective hinge offset. This is further encouraged by centrifugal stiffening during operation.

Since the section consists of a continuous bundle of fibers with uniform compaction, a constant area is demanded. Thus, the thickness and planform distributions cannot both be independently specified. If the dimensions are b and h, we can specify one of them to be, say, linear: $b(x) = mx + b_0$. Then $b(x)h(x) = A$ or $(mx + b_0)\, h = A$, leaving

$$h = \frac{A}{(mx+b_0)}.$$

Since it is clear that a nonlinear shape is required, it is reasonable to suggest that a bending stiffness distribution be the specified quantity, letting both b and h conform as required.

For this study, the (planform x thickness) dimensions were chosen as follows. The root sections of the beam are 4 in x 1.5 in for an area of 6 in$^2$. Sections at the apex are 2 in x 3 in with distribution chosen so that the flapwise moment of inertia increases linearly from root to apex.

Choice of the baseline material was made by creating a qualitative merit factor for each of graphite, S-glass, and Kevlar. In order to make a higher merit factor represent a better choice of material, properties were placed in the numerator if a high value was favorable and in the denominator if a low value was favorable. The favorable properties used were high bending stiffness (E), high ultimate tensile strength ($F_{TU}$), high ratio of bending to shear modulus (E/G), and low specific weight ($\gamma$). Thus, the merit factor becomes

$$M = \frac{F_{TU}E^2}{G\gamma}$$

and $M_{Kevlar}$ = 1.6 million, $M_{graphite}$ = 2.3 million, $M_{S\text{-}glass}$ = 0.2 million. Graphite is chosen as the baseline material.

Nearly all centrifugal force from the blade enters the EPB via bearing in the clevis track at the apex. To keep trapeze effects at a minimum, this region should approximate a perfect apex as nearly as possible, but it cannot be so sharp that bearing allowables are exceeded in the composite. For this study, a 3-inch-diameter racetrack was used giving an approximate bearing area of (3) (1.5 /2) = $\underline{7.07\ in^2}$ and a bearing stress (steady) of (61 kips) (1.5 ult) /($7.07\ in^2$) = $\underline{12.94\ ksi\ ult}$.

It has been noted that the primary size drivers are static droop and hard start-up torque. There are two ways of considering each - shear and moment applied at the apex or shear alone. These represent the two configurations of the EPB. The plain elastic pitch beam (PEPB) has all blade loads (except pitching moment) reacted at the apex - both shears and moments. The classic elastic pitch beam (CEPB) reacts only shears. Moments are "kicked" through the control tube to be manifested as couples - increased shear at the apex and a kick force at the inboard elastomeric snubber.

The original rough sizing was done for the PEPB. For comparison, stresses were computed for a CEPB of the same size and for four smaller sizes whose dimensions were obtained by multiplying the baseline dimensions by "size factors" .85, .75, .65, and .55. These results are shown in Figure 21 along with deflections of a rigid blade tip. The values on the graph in Figure 21 were obtained from the finite element program, but the hand calculation technique will be shown in this section since it is more accurate than the computer code.

It is customary to use multiple -g accelerations applied to the deployed blade mass to model effects of wind gusts on a parked craft, hard landings, and similar phenomena. While values much higher than 3g are often used to design to ultimate strengths, 3g has been used here to design to conservative stress levels far below ultimate. Blade weight, 225.8 lb, was integrated from a dis-

tribution used for this size rotor in the Advanced Rotor Research Study (Reference 3). Its center of gravity is at 56% radius. A 3g droop load, therefore, places a shear of .677 kips and a moment of (.677k) (.56) (372 in) = 141 in-kips at the apex of the EPB. For the CEPB, the moment component reacts as a couple separated by 40 in - the root to apex radial distance. Thus, (141 in-kips)/(40 in) = 3.53 kips added shear at the apex. So the CEPB load is 4.20 kips. Although the moment has been relieved, the shear is much larger. Notice in Figure 21 that there is very little difference in droop mode between CEPB and PEPB. The hand calculation for droop deflection follows.

Since the moment of inertia varies along the span, the beam equation needs to be re-integrated.

$$E\frac{d^2y}{dx^2} = -\frac{M(x)}{I(x)} \tag{1}$$

In general, for a cantilever with tip load, the bending moment along the span will be $V_\ell(\ell-x) + M_\ell$, where $V_\ell$ is the shear applied at the tip, $\ell$ is the length, and $M_\ell$ is the pure moment applied at the tip. (For the CEPB, $M_\ell = 0$.) The linear variation of $I(x)$ is written $I_m x + I_o$, where $I_o$ is the moment of inertia at the root and $I_m$ is the slope of $I(x)$ vs. $x$.

The beam equation is then

$$E\frac{d^2y}{dx^2} = -\frac{V_\ell(\ell-x)+M_\ell}{I_m x+I_o} \tag{2}$$

Integrating once yields the slope along the span:

$$E\frac{dy}{dx} = E\theta = \frac{V_\ell x}{I_m} - \frac{I_m(V_\ell \ell+M_\ell)+V_\ell I_o}{I_m^{\,2}}\ell n\,(I_m x+I_o) + C_1 \tag{3}$$

Integrating again yields the deflection:

$$Ey = \frac{V_\ell x^2}{2I_m} - \frac{V_\ell I_o+I_m(V_\ell \ell+M_\ell)}{I_m^{\,2}}\left[\frac{I_m x+I_o}{I_m}\ell n\,(I_m x+I_o) - x\right] + C_1 x + C_2 \tag{4}$$

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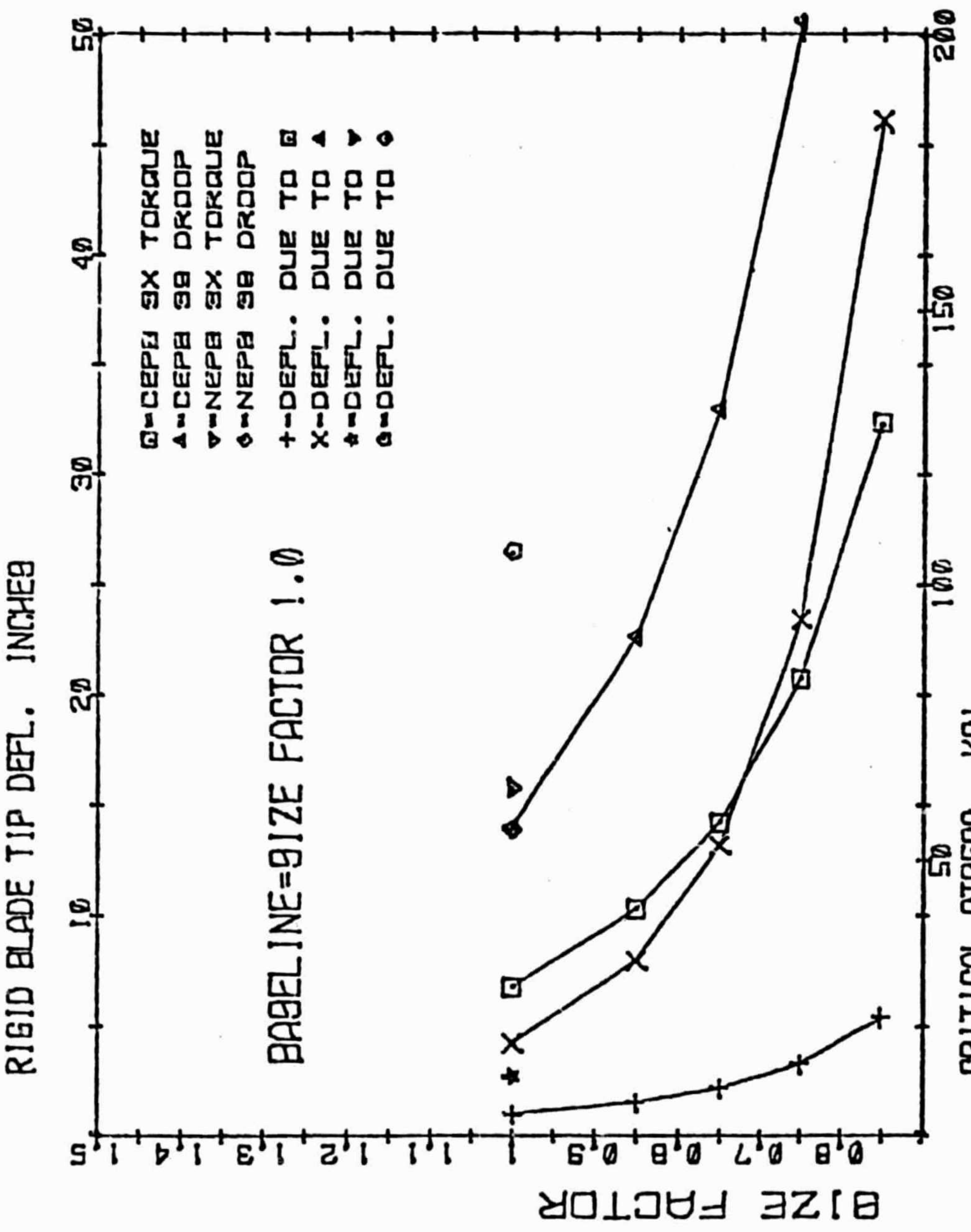


Figure 21. Critical Stresses and Tip Deflections.

Substituting boundary conditions for cantilever yields for the integration constants:

$$C_1 = \frac{I_m(V_\ell \ell + M_\ell) + V_\ell I_o}{I_m^{\,2}} \ell n\ I_o \tag{5}$$

$$C_2 = \frac{V_\ell I_o^{\,2} + I_m I_o(V_\ell \ell + M_\ell)}{I_m^{\,3}} \ell n\ I_o \tag{6}$$

For the present case, $I_o = \frac{1}{12}(4)(1.5^3) = 1.125\ \text{in}^4$,

$\ell = 40$ in., $I_\ell = \frac{1}{12}(2)(3^3) = 4.5\ \text{in.}^4$,

So $I_m = (I_\ell - I_o)/\ell = .0844\ \text{in.}^4/\text{in.}$

For CEPB, $V_\ell = 4.2$ kips, $M_\ell = 0$

$C_1 = 312.58$ psi

$C_2 = 4166.5$ lb/in

$\theta_{apex} = -.0433\text{rad}\ (-2°9')$

$y_{apex} = -1.29"$

$\sigma_{crit} = \frac{M_o c}{I_o} = \frac{V_\ell \ell c}{I_o} = \underline{56\ \text{ksi}}$

Rigid blade tip deflection:
(372in-10in) (1.29in/40in) = <u>11.67in</u>

See Figure 22a

For PEPB, $V_\ell = .677$ kips, $M_\ell = 141$in-kips

$C_1 = 247.04$ psi

$C_2 = 3292.9$ lb/in

$\theta_{apex} = -.0664\text{rad}\ (-3°48')$

$y_{apex} = -1.66"$

$\sigma_{crit} = \frac{M_o c}{I_o} = \frac{(V_\ell \ell + M_\ell)c}{I_o} = \underline{56\ \text{ksi}}$

Rigid blade tip deflection:
(372in-50in) tan (.0664) + 1.66in
= <u>23.07in</u>

See Figure 22b

The "rigid blade tip" is the quantity of interest which embodies the characteristics of the two EPB configurations. Both elastic shape and joint design are represented by this number. Once a particular blade has been defined, its additional sag is merely added.

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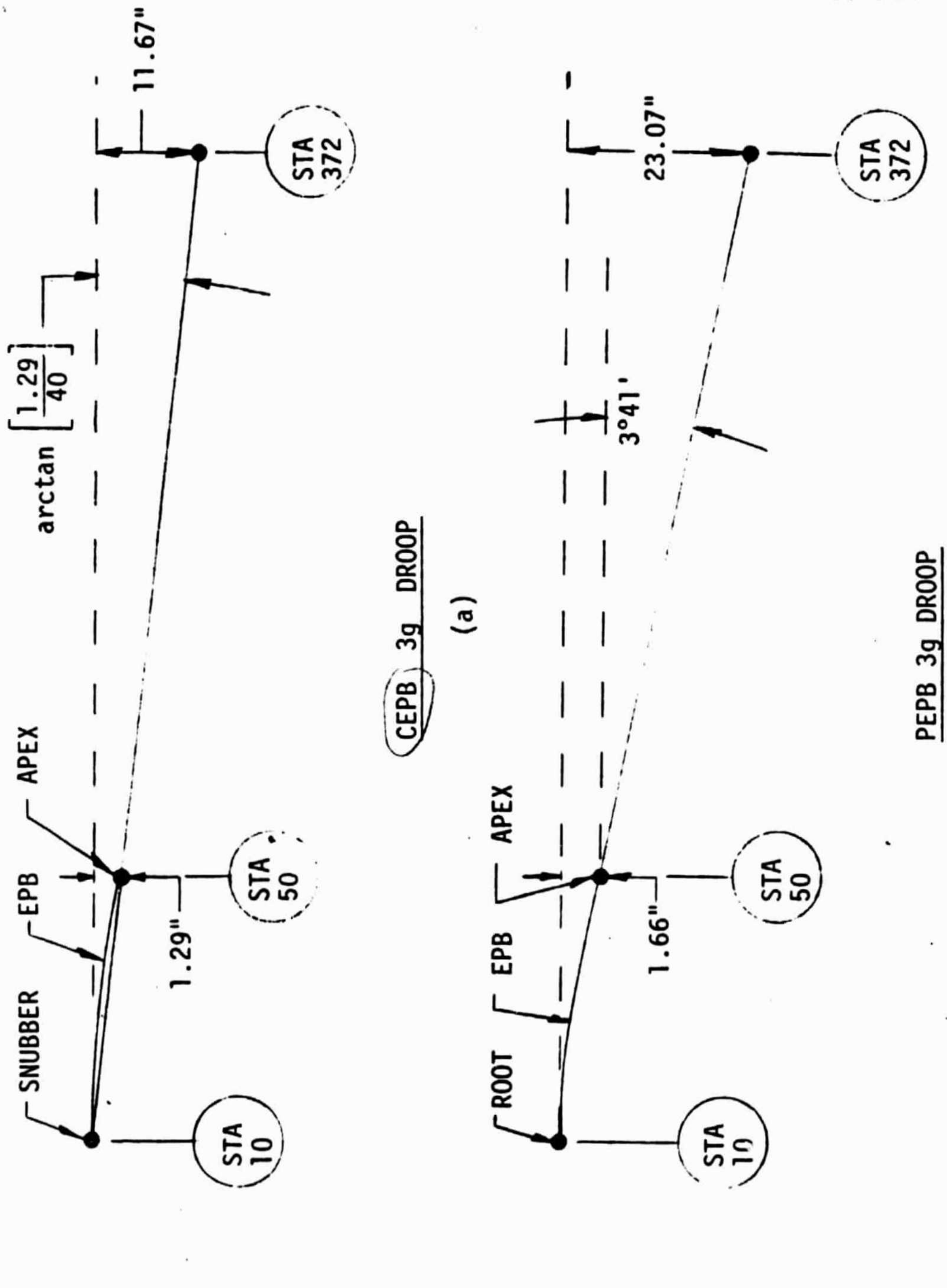


Figure 22. Trigonometric Composition Of Tip Deflections.

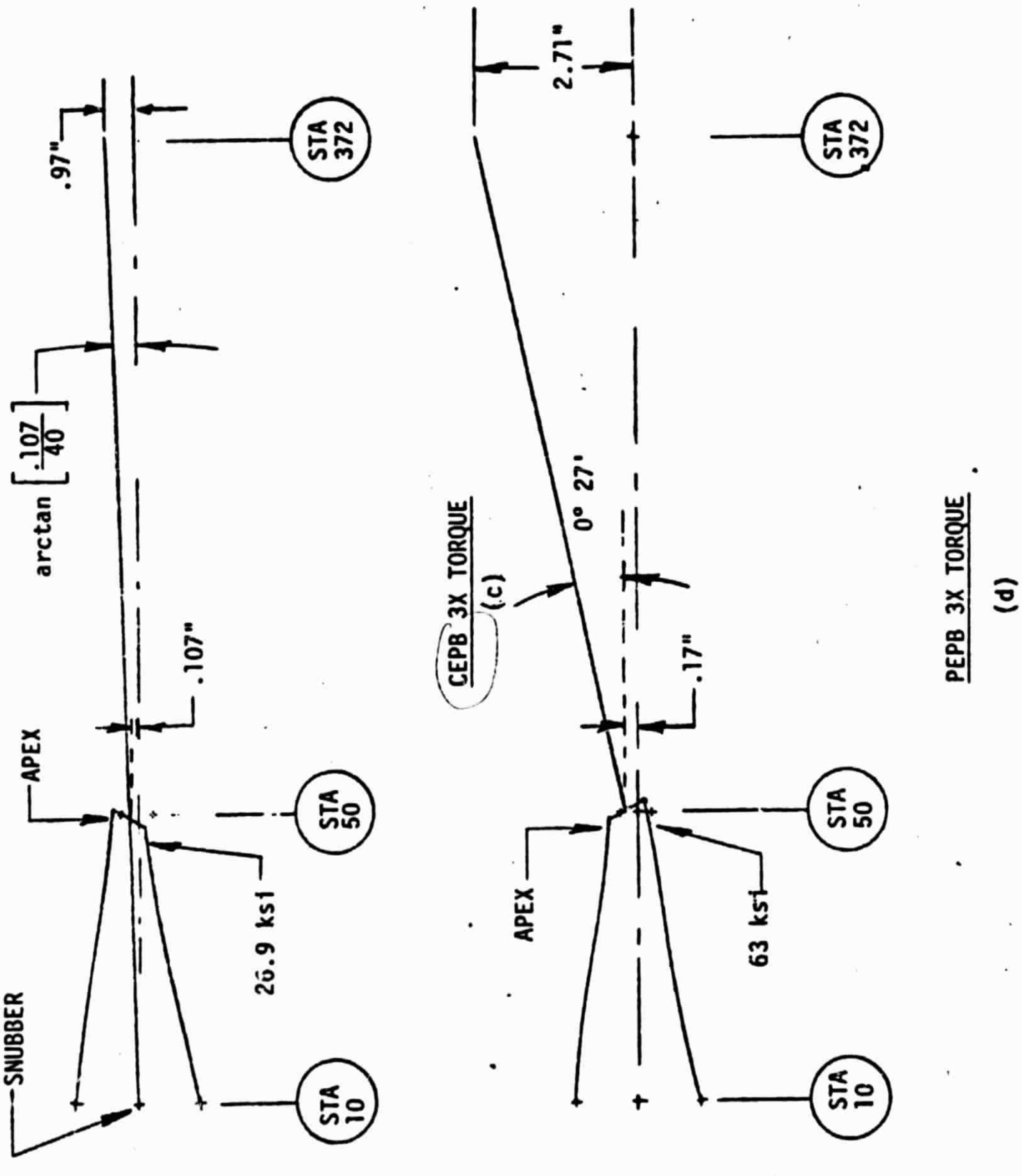


Figure 22. (Continued)

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If the clevis grips the apex and does not allow it to rotate freely as in Figure 22a, the CEPB will behave as a cantilever with guided tip slope. A variable boundary condition such as this complicates the analysis considerably and is beyond the scope of this study. The effect, however, is to further stiffen the EPB in droop.

It is seen that there is no stress benefit of the CEPB in droop mode. Because of the A-frame geometry, this is not the case for in-plane loads. Two completely different regimes of behavior are involved. When in-plane shears are applied to the apex, the structure behaves as a truss with clamped joints. When in-plane moments are applied, no truss action can take place. This mode approximates two nonconnected cantilevers with tip moments.

The critical in-plane condition is hard start-up. This is a sudden torque load applied to the rotor shaft. This value is estimated using a "red line" horsepower for the transmission, an inertial shock load factor of 2, and an ultimate factor of 1.5. Hereafter this is referred to as three times red line torque, or simply 3x torque. Red line horsepower is 2500. Normal speed is 217 rpm. Thus, limit torque is

$$Q = \frac{P}{\Omega} = \frac{(2500\text{hp})(33000\text{ft-lbs-min/hp})}{(217\text{ rpm})(2\pi)} = 60508\text{ ft-lbs}$$

$$Q = 726.1\text{ kip-in}$$

$$3\text{x torque (ult)} = 2178\text{ kip-in or } 544.6\text{ kip-in/blade}$$

Since this is an inertial and aerodynamic source, the torque can be converted to an "equivalent shear" acting at the center of gravity and center of drag of the blade. A good compromise (and conservative) between these two centers is three-quarters radius, or 279 in.

$$V_{equiv} = (544.6\text{ kip-in})/(279\text{ in}) = 1.952\text{ kips}$$

The in-plane apex loadings on the PEPB are then 1.952 kips shear and (1.952k) (279 in - 50 in) = 447 kip-in moment. For the CEPB, it is (1.952k) + (447k /40 in) = 13.13 kips shear.

The critical stresses for this loading (Figure 21) occur at the effective lag hinge near the apex. It is of interest to note that, due to the geometry, the apex rotates in opposite directions for shear and moment loadings in the same directions. In the CEPB, the shear effect of the apex rolling "backward" into the load is seen, although it has no effect on sweeping of the blade, which is governed by the translation of the apex - as was assumed with the drooped CEPB. In the PEPB, the moment effect of the apex rolling forward with the load completely overwhelms the shear effect. Here it does contribute to blade sweep. The rigid blade tip excursions for these cases were computed in a fashion identical to that used for droop. Apex deflections and rotations were obtained from output of the finite element computer code discussed below. See Figure 22c and d.

The CEPB configuration improves the stress situation considerably in the in-plane direction, but, because the droop condition has been shown to be critical, no downsizing of the beam cross-sections is possible.

Enough data exists at this point to make a computation of hub moment stiffness and effective flapping hinge offset. The static effective hinge is thought of as being located at the point of maximum elastic curvature of the pitch beam. The effective dynamic hinge can be quite far outboard of the static hinge.

Hub moment stiffness is a rotational spring constant for the tilt, with respect to the plane of rotation, of a disk defined by the center of rotation and the rigid blade tip. Any "unit" load will do, and so the droop loads above are used. These yield a first-order approximation. Actual values will be slightly stiffer since the droop deflections do not include centrifugal stiffening of the EPB.

The calculation procedure for the two concepts follows.

| CEPB | PEPB |
|---|---|
| Hub Moment: (.6774k)(208.32in)=141 in-kips | Hub Moment: 141 in-kips |
| Disk Tilt: $\arcsin\left[\frac{(11.67in)}{(372in)}\right]$ $= .0314 rad\ (1°48'0)$ | Disk Tilt: $\arcsin\left[\frac{(23.07in)}{(372in)}\right]$ $= .0621 rad\ (3°33')$ |
| Hub Moment Stiffness: | Hub Moment Stiffness: |
| $K_m = \frac{(141 in\text{-}k)}{(.0314 rad)} = 4490$ in-kips/rad | $K_m = \frac{(141 in\text{-}k)}{(.0621 rad)} = 2271$ in-kips/rad |
| $K_m$ = 374,204 ft-lbs/rad/blade | $K_m$ = 189,211 ft-lbs/rad/blade |

Hinge offset: $e = K_m / \left[S_B \Omega^2 (\frac{B}{2})\right]$, where $S_B$ is the first mass moment of the blade, $\Omega$ is the angular frequency of the rotor, and B is the number of blades. Here, $S_B = mr_{cg} = \frac{(224.81 bs)}{(32.2 ft/s^2)}\left(\frac{208.32}{12}\ ft\right) = 121.74$ slug-ft and $\Omega = 22.72$ rad/s (217 rpm). $S_B\ \Omega^2 = 62{,}842$ lbs.

| CEPB | PEPB |
|---|---|
| $e = \frac{(374{,}204 ft\text{-}lbs/rad)}{(62{,}842\ lbs)}$ $= 5.95\ ft = 19.2\%$ | $e = \frac{(2)(189{,}211 ft\text{-}lbs/rad)}{(2)(62{,}842\ lbs)}$ $= 3.01\ ft = 9.71\%$ |

The baseline CEPB is seen to be very stiff with a very large hinge offset. It will be shown below that these quantities can be tailored to any value desired well below the design goal specifications of 172,500 ft-lbs/rad and 5% hinge offset.

5.1.4 Overview Of Analysis Routine. The computer code chosen for the analysis is a simplified finite beam element routine. The structure is divided into a number of finite length beams of arbitrary cross-section and properties. A stiffness matrix for the structure is generated by the code from the input data given. This is used to construct a set of simultaneous equations for each load case requested. Their solution yields the behavior of the arbitrarily complex structure.

There are four limitations to the program which need to be noted.

(1) The static beam bending problem is solved - no dynamic effects are modeled.

(2) The tension beam equation is not used - centrifugal stiffening cannot be modeled directly. The stiffened solution can be obtained by using the program in an iterative fashion. This process is lengthy, however, and is beyond the scope of the project.

(3) Beam shear is not treated - length of beam elements cannot be decreased indiscriminately. Each finite element must be long enough to ensure that it behaves in the bending regime.

(4) Second-order effects due to large deflections are not included.

5.1.5 Laminated Elastomer Concept. The laminated elastomer concept consists of splitting the beam cross-section into a series of stacked thinner leaves bonded together by intervening layers of an elastomeric compound. (The type of elastomer is yet to be defined.) When the "h" term in the moment of inertia is divided into a number of smaller sections, the beam is softened. This is because $(\Sigma h_i)^3 > \Sigma(h_i^3)$. Tractive forces supplied by the elastomer along the bond surface tend to stiffen the sandwich but are negligible to fourth order, since $\frac{G_{rubber}}{E_{composite}} \approx 10^{-4}$. If end fixtures are present which prevent relative shearing of the leaves - as is the case with the blade attachment clevises - a unique property results. When the beam is loaded in the collective mode (Figure 23a), it behaves as the full thickness member would because of the end fixity. But when the beam is loaded in the cyclic mode (Figure 23b), positive curvature on one side is balanced by negative curvature on the opposite side. This allows the soft mode bending to occur.

This dual personality is ideal for helicopter applications. The collective or "hard mode" bending corresponds primarily to static droop, for which great stiffness is desired. The cyclic or "soft mode" bending corresponds to dynamic flight loads, for which tailorable softness and hinge offset is desirable.

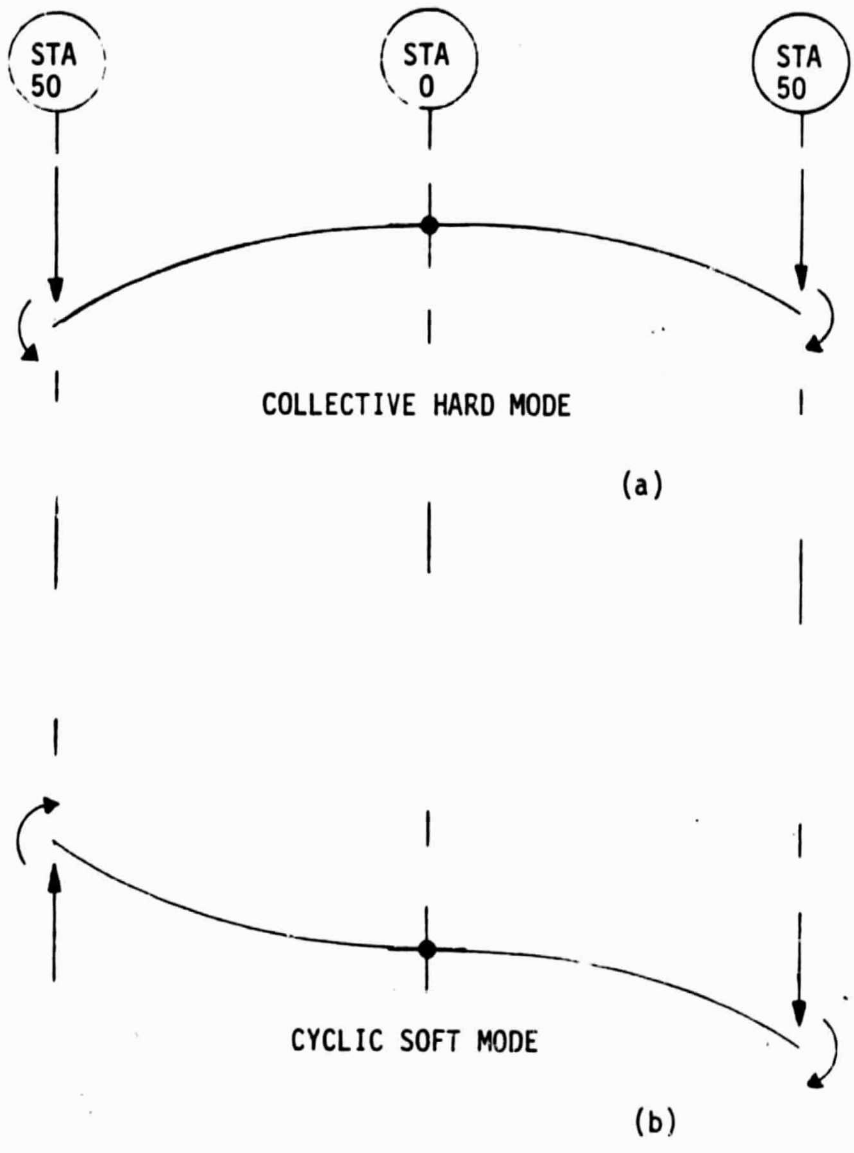




Figure 23. Bending Modes Of Laminated Beam With Clamped Ends.

This softening is quite dramatic. If the beam "H" is divided into n small leaves each of height "h" such that

$$\sum_{i=1}^{n} h_i = H,$$

then each leaf is $\frac{1}{n^3}$ as stiff and there are n leaves, so the resulting beam is $\frac{1}{n^2}$ as stiff as the unleaved beam. Torsion is softened by a similar mechanism.

Figure 24 shows the cross-sections of the leaved beams considered in this study. Figure 25 is a plot of the elastic properties versus number of leaves.

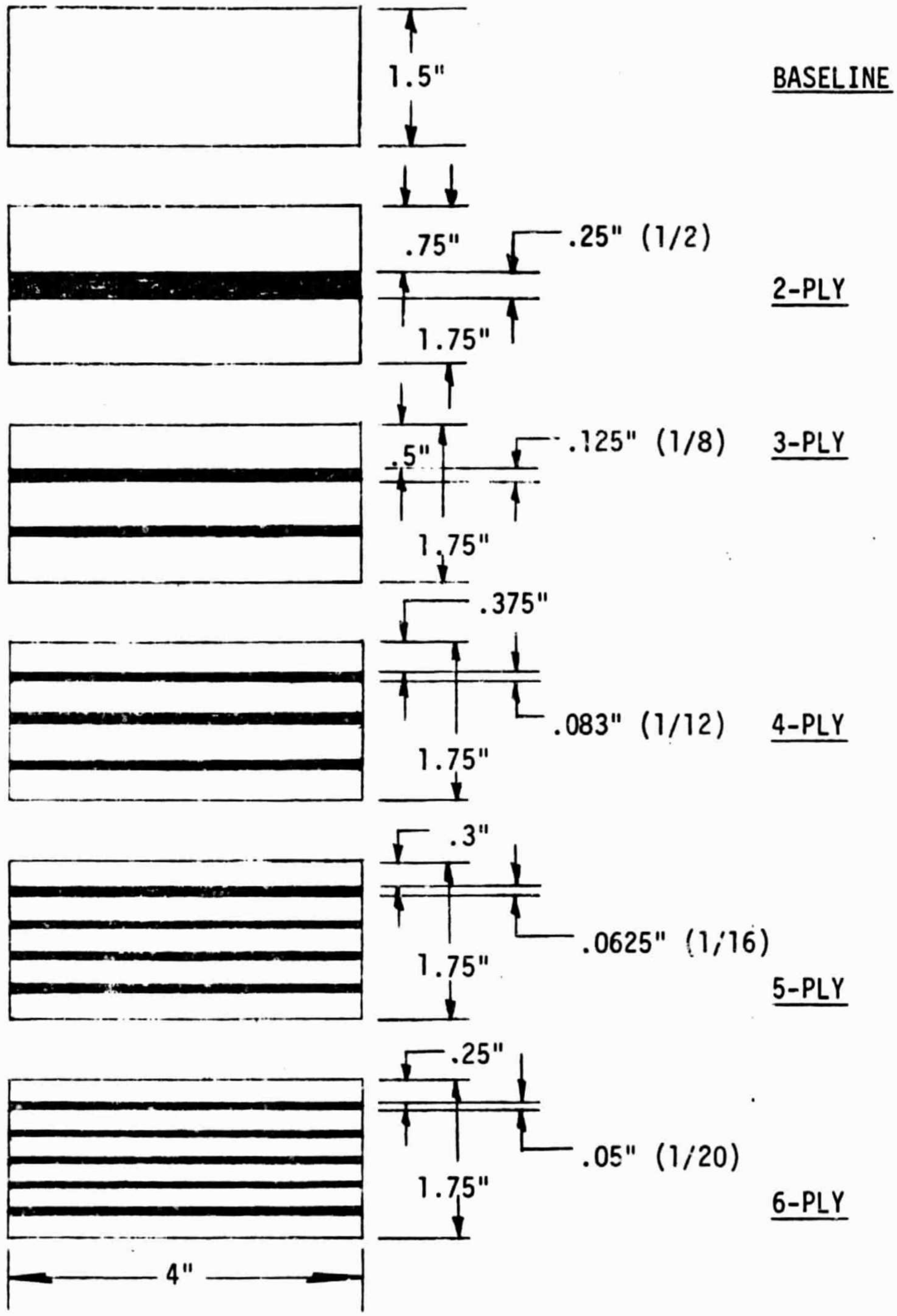


Figure 24. Laminated Elastomer Cross-Sections.

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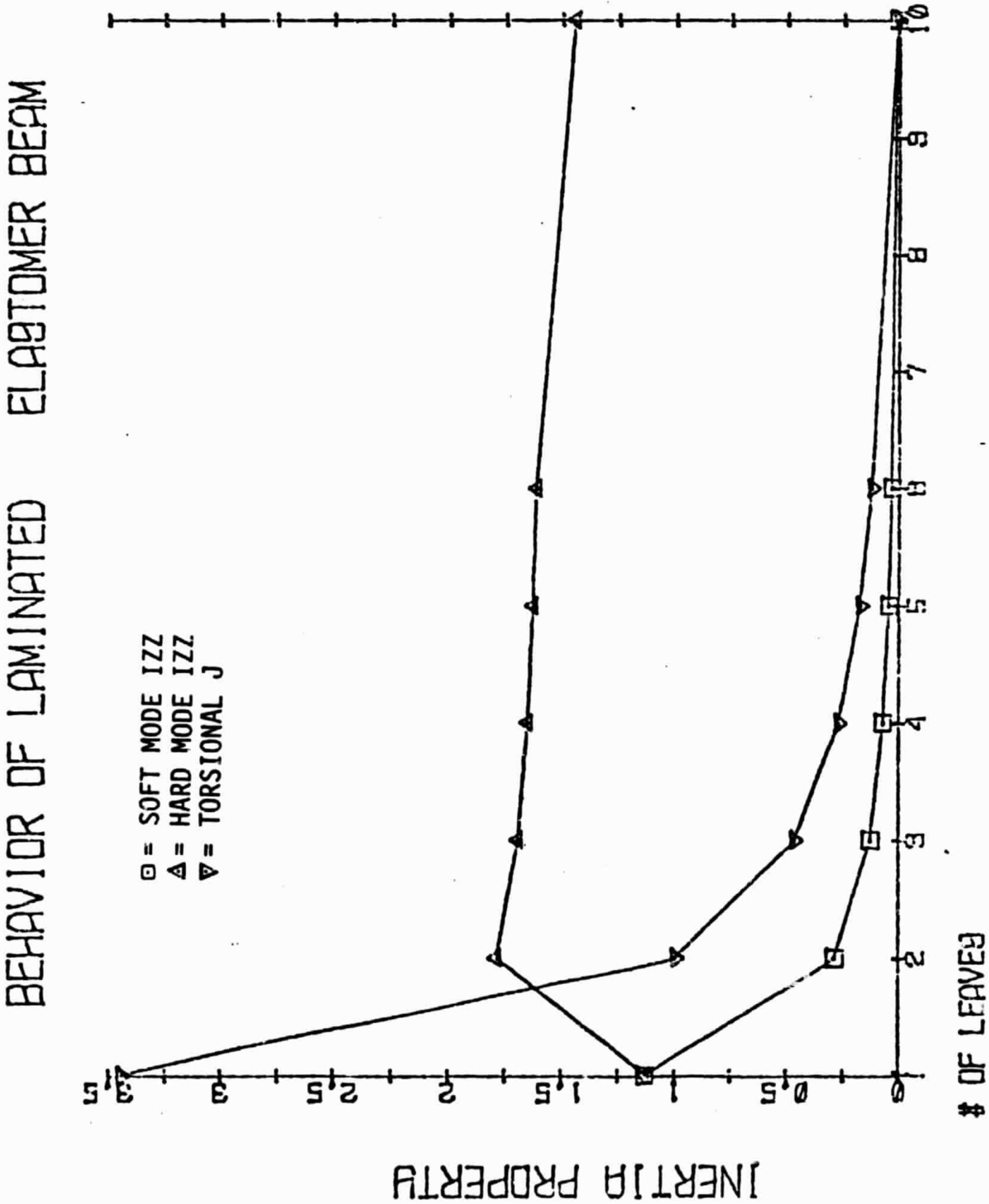



Figure 25. Elastic Properties Of Laminated Beam.

The hub moment stiffness can now be computed for the elastomer sandwich beams to show the wide range available for tailoring. The 2-ply and 5-ply examples are shown.

**2-Ply**

| CEPB | PEPB |
|---|---|
| Hub Moment Stiffness: | |
| $K_m$ = 82,572 ft-lbs/rad | $K_m$ = 48,715 ft-lbs/rad |
| Hinge Offset: | |
| e = 4.24% | e = 2.50% |

**5-Ply**

| CEPB | PEPB |
|---|---|
| $K_m$ = 39,736 ft-lbs/rad | $K_m$ = 26,446 ft-lbs/rad |
| e = 2.04% | e = 1.36% |

5.1.6 Torsional Behavior Of A-Frame. The torsional behavior of the A-frame portion of the EPB structure is independent of the classic and plain configurations, and therefore no distinction will be made in this section.

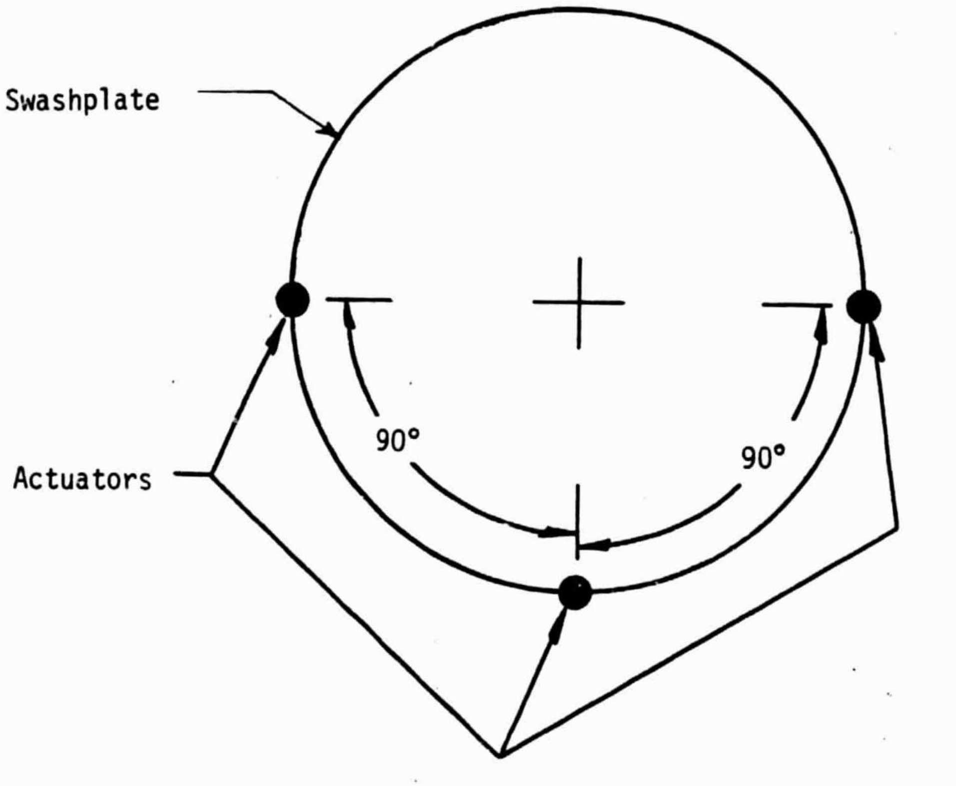
Bearingless rotors have an inherent torsional stiffness which can be problematic when adapting them to existing control systems. The flex member for pitch must also accommodate all bending loads with acceptably low deflections. This requirement, if stated naively, results in a section far too stiff to be usefully pitched for helicopter applications. Advanced uniaxial composites with a high E to G ratio make the concept feasible, but alone are not enough to ensure a successful design. The I to J ratio of the section must also be tailored to a sufficiently high value. A very eccentric rectangle has a much lower J than a square, even though they both enclose the same area. Conversely, a very high I can be achieved with thin rectangles by moving far

above and below the neutral bending axis, as is done in an I-beam. Since the helicopter rotor must be stiff in bending about both axes, an ideal shape for the flex beam is, in principle, a pseudo-cruciform made from two I-beams crossed at their centroids at right angles. The three primary attributes - vertical and horizontal web depth and material thickness - can be uniquely defined for any desired combination of the three variables $I_{flat}$, $I_{edge}$, and J.

The manufacturing complexity that would be involved in building such a cruciform with spanwise varied attributes makes it much less attractive. Notch effects at the re-entrant corners could also affect fatigue life. A relatively simple way to achieve the same result is to use the leaved elastomer section discussed above. It was introduced for another purpose -- the added torsional softness is a side benefit.

The primary purpose here is to keep control loads required for full stacked travel within current RSRA capability. The RSRA has three actuators on the swashplate. Each is a double-stage actuator capable of 4800 lbs. limit load per stage. In normal single-stage operation, the second stage remains as a redundant backup unit. If necessary, both stages can be linked for 9600 lbs. limit load, but with loss of redundancy. While Kaman feels that redundancy should not be sacrificed, both single- and double-stage capabilities are shown for perspective. Because of the arrangement of the actuators on the swashplate (Figure 26), a worst-case condition is with two actuators doing all the work. This corresponds to a maximum swashplate up-thrust of 9600 lbs. single-stage and 19,200 lbs. double-stage or 2400 lbs. and 4800 lbs. available per blade, respectively. With the 15.75-in. pitch horn, this is a control torque of 37.8 in-kips and 75.6 in-kips respectively.

The triangular geometry forces the torsion of the structure to be more complex than the bending modes. In order to conform to the boundary conditions, each leg must bend about its flapwise axis as well as twist. Edgewise bending is, in principle, present also but is a cosine effect and is negligible to third order. It is similar in shape to a double cantilever curve with two inflection



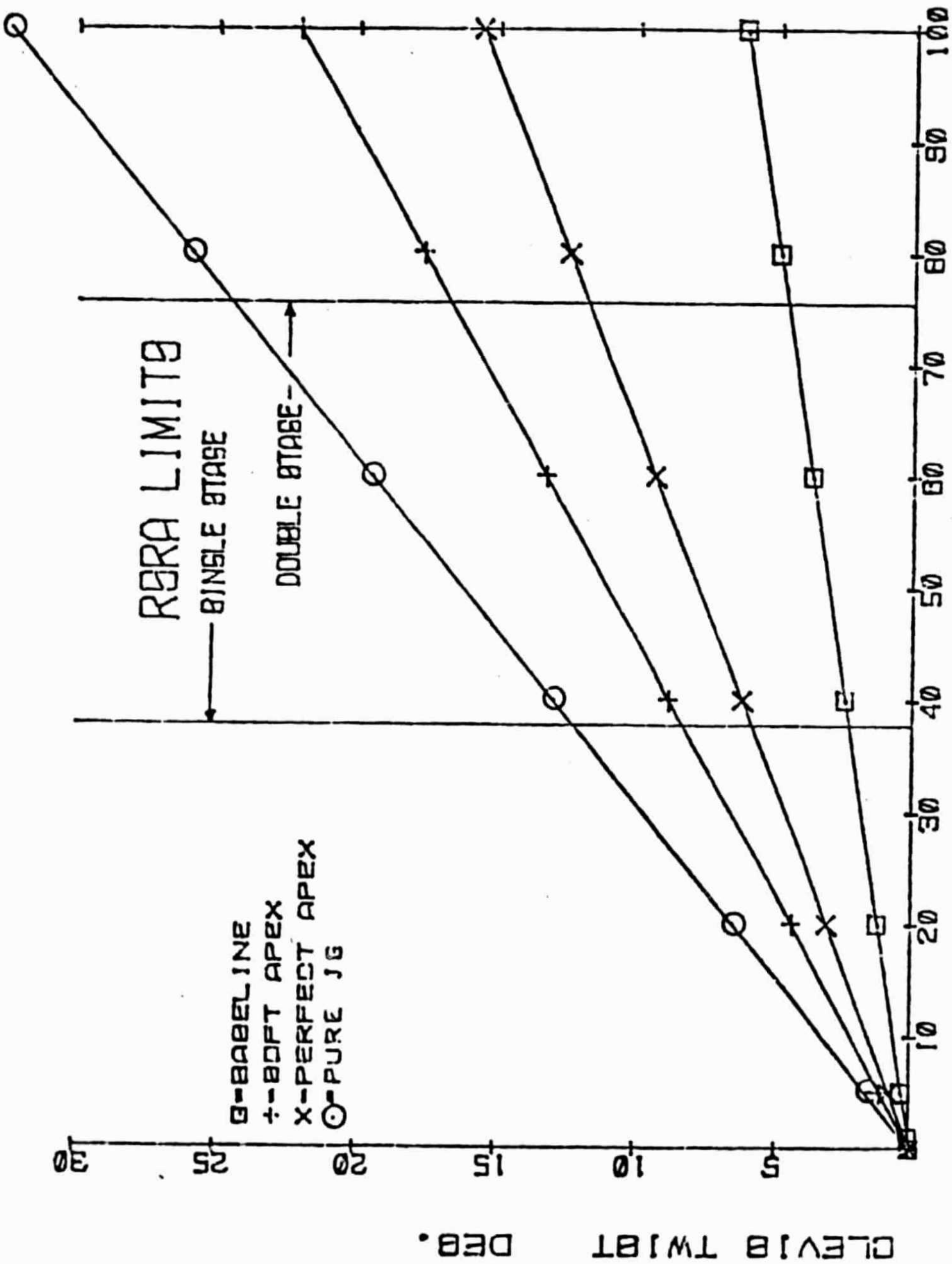

(Not intended to imply any absolute azimuthal orientation)

Figure 26. RSRA: Relative Location of Actuators.

points and a point of maximum bending moment somewhere about mid-span. As the finite offset at the clevis increases, the maximum bending moment migrates outboard until, in the limit, it is located at the clevis when the two legs are parallel. Since, in the tailored EPB, flapwise stiffness is greatest near the clevis, it is clear that the better the clevis approximates a perfect apex, the softer will be the torsional character of the structure. The behavior of this bending term is demonstrated by four cases seen on the graphs in Figure 27. The "baseline" case represents the baseline EPB as it really is. The "pure JG assumption" is a hand calculation neglecting the presence of the bending term. This case demonstrates the importance of the bending term in stiffening the actual structure. The "perfect apex" case demonstrates the softening effect of chasing the maximum bending moment inboard. The "retailored" case keeps the true clevis and max bending moment at the baseline position but shows a similar softening effect by maintaining the root dimensions outboard to this point. The "soft apex" case is similar to the pure JG assumption. By connecting the apex with a nodal element of the same properties as the EPB tip (rather than an infinitely stiff clevis), more deformation is allowed to flow into twisting the bars with less forced bending.

It is seen that the baseline EPB is far too stiff for the RSRA controls to supply full stacked travel which could require an extreme range of about 40°. Half that range can be eliminated by prebiasing the elastic rest position of the clevis in the middle of the range (about +20°).

Application of the laminated elastomer concept brings the remainder of the necessary softening to the design. From examination of the four cases discussed above, it can be observed that the pseudo-cruciform shape, while allowing order of magnitude reductions of J, would be disappointing in the EPB application. This is because it retains the high flapwise I - indeed that is its purpose - and the EPB clevis forces a significant bending contribution. The leaved elastomers provide a practical way around this problem. The bending induced by collective pitch torsion of the A-frame is soft mode bending. The effects of using a two-ply and a five-ply EPB are shown in Figure 27b. Superposition of a cyclic pitch input stiffens the system somewhat. The effective




(a)

Figure 27. Torsional Stiffness Of Elastic Pitch Beams.

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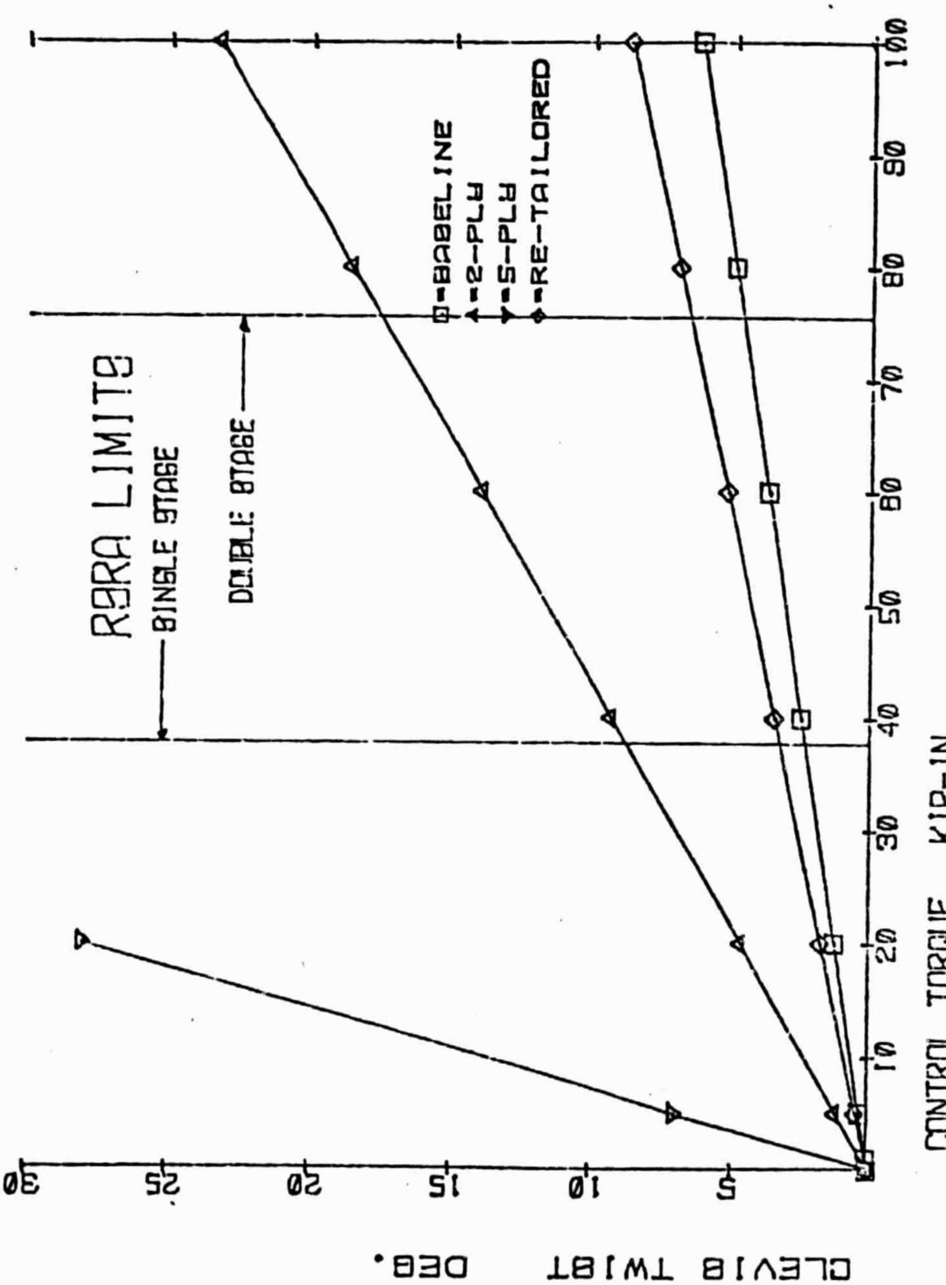


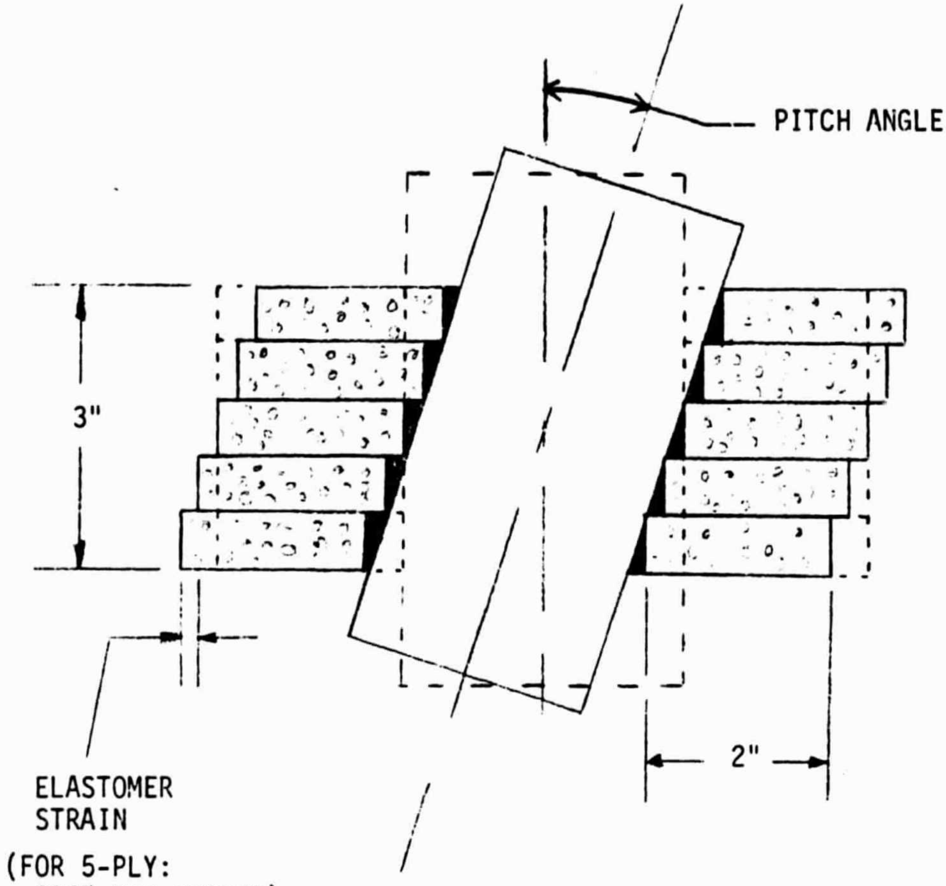

(b)

Figure 27. (Continued)

EI of the induced bending is a function of the ratio of the differential pitch of the two opposing blades to the collective pitch. In simpler words, it depends on ratio of the difference of two numbers to their average. The relationship is very likely nonlinear and would require a NASTRAN or similar analysis, but limits can be defined. Pure collective input is the soft limiting case. Pure cyclic input is the worst case. Since worst case input must occur during autorotative maneuvers, this is obviously an area requiring more detailed work.

There is, however, much confidence that further detailing of the clevis design can restore most of this softness by allowing elastomer shear in the direction tangential to the "circle" traced by the clevis during operation. Due to the small moment arm, 3 in, large pitch excursions are possible with very little elastomer strain - further, there is no centrifugal stiffening of this spring mode. See Figure 28. Notice that this is not a redesign of the elastomer system. It is a natural mode of deformation of such a stacked beam. The foregoing analysis merely made an assumption that the clevis fixture was designed to prevent it. It might just as easily be detailed to allow it. Notice also this is not a double working of a given elastomer region. Radial slip through the hub center (that mode invoked for cyclic flapping softness) is confined to the most inboard regions while the tangential shearing is a phenomenon primarily local to the clevis region.

5.1.7 Structural Coupling. Under high speed, high thrust conditions, certain aerodynamic effects provide significant contributions to the damping of undesirable blade motions. Some of these effects can be induced by providing elastic coupling in the flex beam. Elastic coupling means that the inertia tensor is oriented such that bending takes place about nonprincipal axes. That is, for example, that a load applied in the up-down plane produces bending deflection in both the up-down and the left-right planes. It is mandatory to be aware of the coupling characteristics of the flex beam structure, since a coupled deflection in the wrong direction can be destabilizing. Because of normal manufacturing tolerances, it is always possible that a small amount of unintended destabilizing coupling could occur unless proper coupling behavior




SECTION THROUGH CLEVIS LOOKING INBOARD

FIVE-PLY BEAM SHOWN

Figure 28. Tangential Shear Mode In Pitch.

is understood and is intentionally designed into the system. The stabilizing coupling which is to be enhanced for the purposes of this study is lag-pitch and lag-flap, that is, any input of lag deflection should also produce a nose-up pitch and flap up.

With the elastic pitch beam there are many ways of inducing coupling. Seven possibilities were investigated here:

(1) Baseline - "BAS" - Theoretically Uncoupled Case (Figures 29a and 31)
(2) Base Legs Offset - "BOFF" - Vertical Offset Of Root Fixtures (Figures 29b and 32)
(3) Synchronous Base Twist - "BTWSYNC" - Same Direction Twist Of Root Fixtures (Figures 29c and 33)
(4) Opposable Base Twist - "BTWSOPP" - Opposite Direction Twist Of Root Fixtures (Figures 29d and 34)
(5) Tip Twist - "TTWS" - Twist Of Apex Racetrack Relative To Clevis (Figures 29e and 35)
(6) Non-Isosceles Planform - "XISOSC" - EPB Legs Are Of Unequal Length Forming A Non-Isosceles Triangle (Figures 29f and 36)
(7) Nonsymmetric Tailoring - "XSYMM" - EPB Legs Are Of Uneven Cross-Section Tailoring (Figures 29g and 37)

Using a vector of the six possible unit loads (unit load being 10 kips or 10 in-kips), a coupling matrix was produced for each case to display the relative effectiveness of each variation. The six unit loads and their sign conventions were defined as follows (analysis code uses a left-handed coordinate system):

| | | | |
|---|---|---|---|
| (F1) | Out-of-plane Shear (CEPB and PEPB) | (10 kips) | (+tending to cone) |
| (F2) | Out-of-plane Moment (PEPB Only) | (10in-kips) | (+tending to droop) |
| (F3) | In-plane Shear (CEPB and PEPB) | (10 kips) | (+tending to lag) |
| (F4) | In-plane Moment (PEPB Only) | (10in-kips) | (+tending to lag) |
| (F5) | Axial Force (CF) | (10 kips) | (+tending to stretch) |
| (F6) | Pitching Moment | (10in-kips) | (+tending to nose-down) |

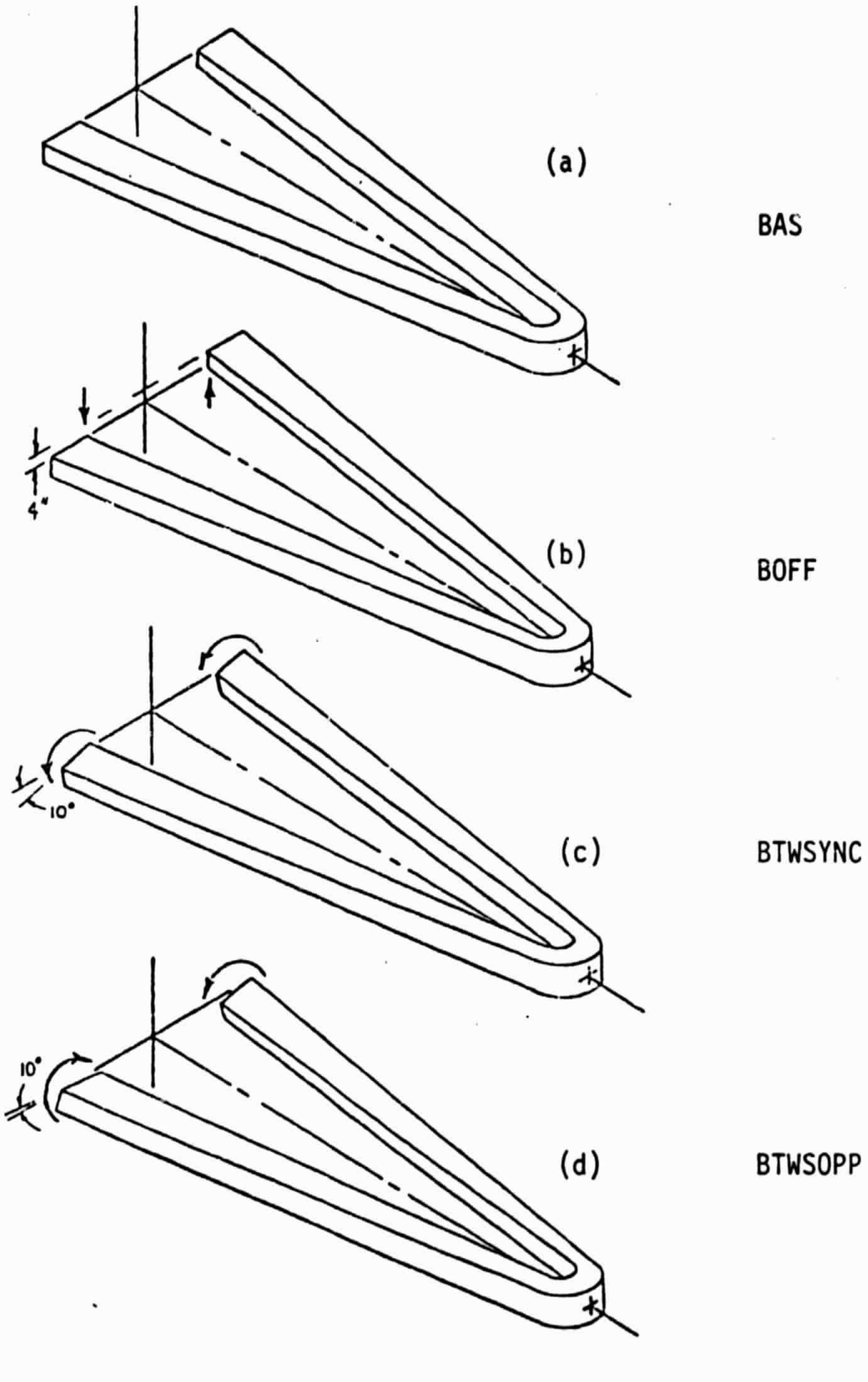
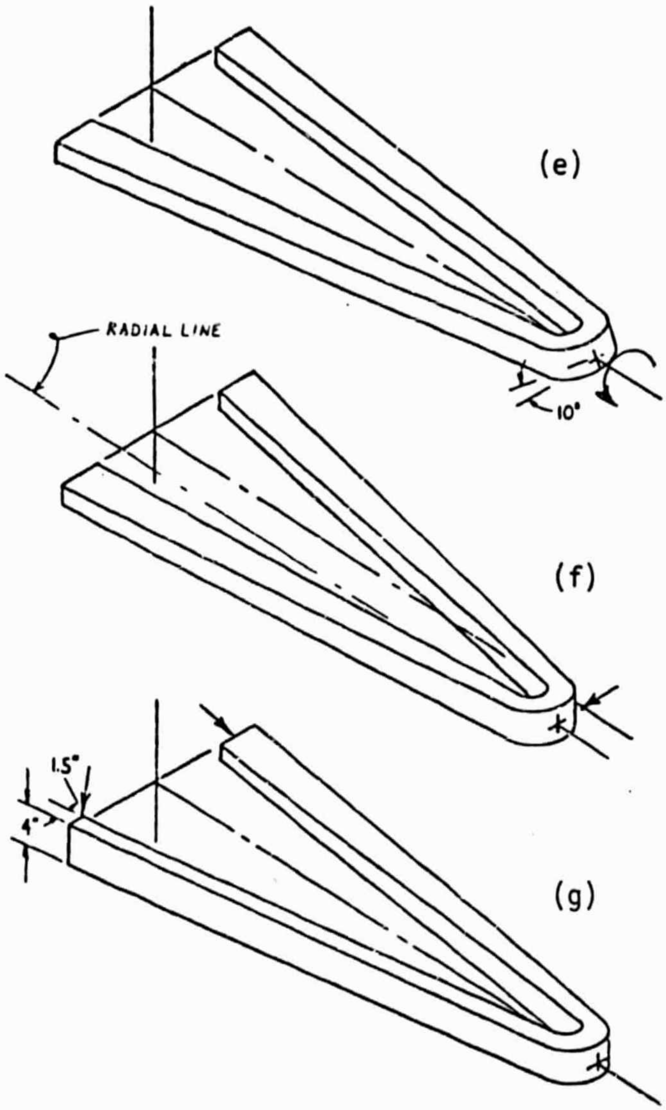



Figure 29. Elastic Coupling Variations.




(e) TTWS

(f) XISOSC

(g) XSYMM

Figure 29 (Continued)

The corresponding deflection vector is then $\delta_1$ = out-of-plane shear type deflection (in); $\delta_2$ = out-of-plane moment type deflection (rad); $\delta_3$ = in-plane shear type deflection (in); $\delta_4$ = in-plane moment type deflection (rad); $\delta_5$ = radial stretch (in); $\delta_6$ twist (rad).

The coupling matrix is then defined by

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} & C_{15} & C_{16} \\ C_{21} & C_{22} & C_{23} & C_{24} & C_{25} & C_{26} \\ C_{31} & C_{32} & C_{33} & C_{34} & C_{35} & C_{36} \\ C_{41} & C_{42} & C_{43} & C_{44} & C_{45} & C_{46} \\ C_{51} & C_{52} & C_{53} & C_{54} & C_{55} & C_{56} \\ C_{61} & C_{62} & C_{63} & C_{64} & C_{65} & C_{66} \end{bmatrix} \begin{Bmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \\ F_5 \\ F_6 \end{Bmatrix} = \begin{Bmatrix} \delta_1 \\ \delta_2 \\ \delta_3 \\ \delta_4 \\ \delta_5 \\ \delta_6 \end{Bmatrix}$$

where for any given i, $F_i$ = unit load <u>and $F_j = 0$ for $j \neq i$</u> (since linear superposition is not valid for large deflections).

The coupling matrices for each variation can be obtained from the finite element code and are shown in Figure 30. The coupling plots for unit loads F1 and F3 are shown in Figures 31 through 37. In these plots. normalized nodal deflections are shown. The normalization is described below. In each of the three views, the two deflection arrays visible in that view are scanned for maximum value. That maximum value is scaled down or up (ie, normalized) to some "arbitrary graph unit" - that being some separation on the paper which is convenient and "comfortable" for the eye to view. The deflection of all other nodes in the view are multiplied by the same normalization factor. In the interest of comfortable viewing, a different size graph unit was chosen for each of the different views because, in some cases, the lead-lag deflections and curvatures would be lost in comparison to the flap deflections. Beam connections between the nodal points are spline fit curves determined by the nodal deflections and rotations at the end points. In the head-on view the cross-sectional shapes of the EPB legs are shown at each nodal loaction. It is in this view that lead-lag deflections can be seen normalized to the

(a) BAS

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 0 | 4 | 1.5 | 0 |

$$\begin{bmatrix} 3.899E-01 & -.128E-01 & 0 & 0 & 0 & -.560E-04 \\ -.128E-01 & 4.957E-04 & 0 & 0 & 0 & 1.846E-06 \\ 0 & 0 & 1.021E-02 & -.415E-03 & 1.470E-08 & 0 \\ 0 & 0 & -.415E-03 & 3.976E-05 & -.107E-08 & 0 \\ 0 & 0 & -.500E-06 & 5.000E-09 & 1.778E-04 & 0 \\ -.875E-04 & 2.870E-06 & 0 & 0 & 0 & 1.086E-03 \end{bmatrix}$$

(b) BOFF

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 0 | 4 | 1.5 | 0 |

$$\begin{bmatrix} 3.647E-01 & -.120E-01 & 7.287E-02 & 2.516E-03 & 2.315E-05 & -.177E-03 \\ -.120E-01 & 4.689E-04 & -.236E-02 & -.843E-04 & -.699E-06 & 5.085E-06 \\ 7.286E-02 & -.236E-02 & 2.496E-02 & 7.804E-05 & 1.292E-05 & -.102E-03 \\ 2.517E-03 & -.844E-04 & 7.832E-05 & 5.770E-05 & -.436E-06 & 3.524E-06 \\ 6.000E-06 & -.500E-07 & 9.400E-06 & -.750E-06 & 1.948E-04 & -.132E-03 \\ -.360E-04 & 5.227E-07 & -.739E-04 & 4.475E-06 & -.132E-03 & 1.060E-03 \end{bmatrix}$$

Figure 31. Coupling Matrices For Variations

(c) BTWSYNC

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0.1745 |
| 10 | 3.414 | 1.811 | 0.1454 |
| 20 | 2.569 | 2.361 | 0.08727 |
| 30 | 2.155 | 2.8 | 0.02909 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 2.155 | 2.8 | 0.02909 |
| 20 | 2.569 | 2.361 | 0.08727 |
| 10 | 3.414 | 1.811 | 0.1454 |
| 0 | 4 | 1.5 | 0.1745 |

$$\begin{bmatrix} 3.780E-01 & -.125E-01 & 1.018E-02 & -.577E-03 & 7.900E-08 & -.780E-04 \\ -.125E-01 & 4.871E-04 & -.294E-03 & 1.661E-05 & 2.316E-09 & 2.385E-06 \\ 1.017E-02 & -.294E-03 & 1.063E-02 & -.439E-03 & 5.699E-07 & -.210E-05 \\ -.576E-03 & 1.660E-05 & -.438E-03 & 4.112E-05 & -.228E-07 & 1.258E-07 \\ -.150E-05 & 3.000E-08 & -.100E-05 & -.150E-06 & 1.778E-04 & 4.022E-07 \\ -.152E-03 & 4.842E-06 & -.440E-05 & 2.483E-07 & 4.282E-07 & 1.072E-03 \end{bmatrix}$$

(d) BTWSOPP

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0.1745 |
| 10 | 3.414 | 1.811 | 0.1454 |
| 20 | 2.569 | 2.361 | 0.08727 |
| 30 | 2.155 | 2.8 | 0.02909 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 2.155 | 2.8 | -0.02909 |
| 20 | 2.569 | 2.361 | -0.08727 |
| 10 | 3.414 | 1.811 | -0.1454 |
| 0 | 4 | 1.5 | -0.1745 |

$$\begin{bmatrix} 3.677E-01 & -.122E-01 & 4.295E-05 & -.247E-05 & -.195E-04 & -.780E-04 \\ -.122E-01 & 4.785E-04 & -.129E-05 & 7.664E-08 & 5.614E-07 & 2.442E-06 \\ 3.196E-05 & -.824E-06 & 1.038E-02 & -.425E-03 & 9.996E-07 & -.220E-03 \\ -.196E-05 & 5.645E-08 & -.424E-03 & 4.033E-05 & -.354E-07 & 1.255E-05 \\ -.191E-04 & 5.392E-07 & 1.500E-06 & 7.000E-08 & 1.778E-04 & 3.500E-08 \\ -.139E-03 & 4.470E-06 & -.219E-03 & 1.252E-05 & 2.735E-09 & 1.077E-03 \end{bmatrix}$$

Figure 31. Continued

(e) TTWS

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0 |
| 10 | 3.414 | 1.811 | 0.02909 |
| 20 | 2.569 | 2.361 | 0.08727 |
| 30 | 2.155 | 2.8 | 0.1454 |
| 40 | 2 | 3 | 0.1745 |
| 40 | 2 | 3 | 0.1745 |
| 30 | 2.155 | 2.8 | 0.1454 |
| 20 | 2.569 | 2.361 | 0.08727 |
| 10 | 3.414 | 1.811 | 0.02909 |
| 0 | 4 | 1.5 | 0 |

$$\begin{bmatrix} 4.766E-01 & -.156E-01 & 1.025E-03 & -.599E-04 & -.151E-05 & 1.995E-04 \\ -.156E-01 & 5.875E-04 & -.101E-03 & 6.609E-06 & 4.240E-08 & -.648E-05 \\ 9.238E-04 & -.972E-04 & 1.017E-02 & -.412E-03 & -.662E-07 & 8.081E-07 \\ -.559E-04 & 6.483E-06 & -.412E-03 & 3.950E-05 & -.169E-07 & -.406E-07 \\ -.800E-06 & 1.000E-08 & 5.000E-07 & -.350E-07 & 1.779E-04 & -.295E-06 \\ 2.384E-04 & -.770E-05 & 5.152E-07 & -.254E-07 & -.285E-06 & 1.100E-03 \end{bmatrix}$$

(f) XISOSC

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 0 | 4 | 1.5 | 0 |

$$\begin{bmatrix} 4.220E-01 & -.139E-01 & 0 & 0 & 0 & 1.500E-05 \\ -.139E-01 & 5.363E-04 & 0 & 0 & 0 & 8.454E-05 \\ 0 & 0 & 1.039E-02 & -.394E-03 & -.752E-03 & 0 \\ 0 & 0 & -.394E-03 & 4.167E-05 & 3.029E-05 & 0 \\ 0 & 0 & -.754E-03 & 3.030E-05 & 2.326E-04 & 0 \\ -.538E-04 & 8.685E-05 & 0 & 0 & 0 & 1.269E-03 \end{bmatrix}$$

Figure 31. Continued

(g) XSYMM

| BEAM STATION | BASE | HEIGHT | ORIENTATION |
|---|---|---|---|
| 0 | 4 | 1.5 | 0 |
| 10 | 3.414 | 1.811 | 0 |
| 20 | 2.569 | 2.361 | 0 |
| 30 | 2.155 | 2.8 | 0 |
| 40 | 2 | 3 | 0 |
| 40 | 2 | 3 | 0 |
| 30 | 1.917 | 3.136 | 0 |
| 20 | 1.75 | 3.436 | 0 |
| 10 | 1.583 | 3.8 | 0 |
| 0 | 1.5 | 4 | 0 |

$$\begin{bmatrix} 1.417E-01 & -.509E-02 & 0 & 0 & 0 & -.326E-02 \\ -.509E-02 & 2.474E-04 & 0 & 0 & 0 & 7.282E-05 \\ 0 & 0 & 1.260E-02 & -.556E-03 & 4.470E-06 & 0 \\ 0 & 0 & -.557E-03 & 4.819E-05 & -.260E-06 & 0 \\ 0 & 0 & 4.500E-06 & -.450E-06 & 1.778E-04 & 0 \\ -.327E-02 & 7.313E-05 & 0 & 0 & 0 & 9.500E-04 \end{bmatrix}$$

Figure 31. Continued

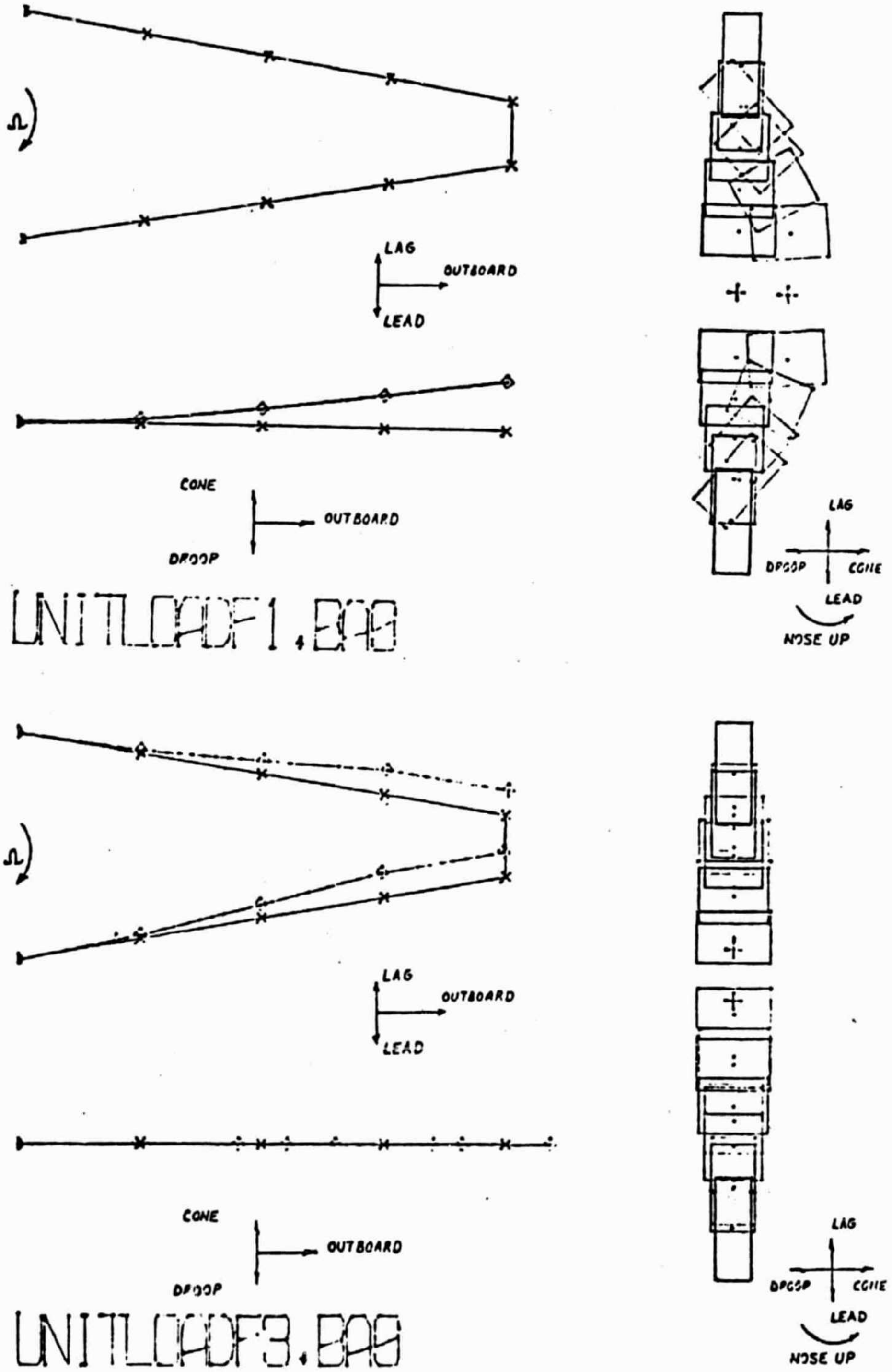



Figure 31. Baseline - Normalized Nodal Deflections.

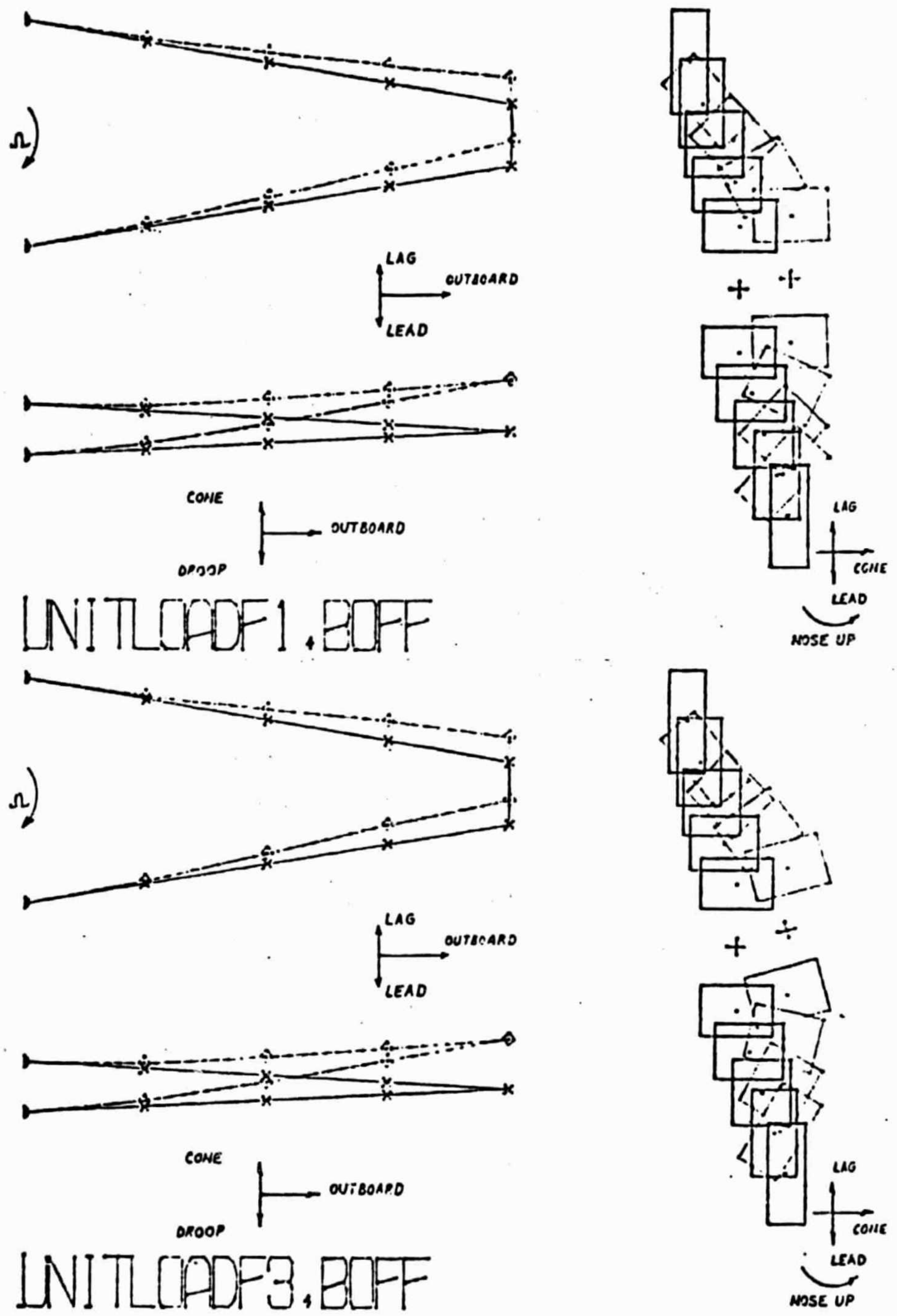



Figure 32. Base Offset - Normalized Nodal Deflections.

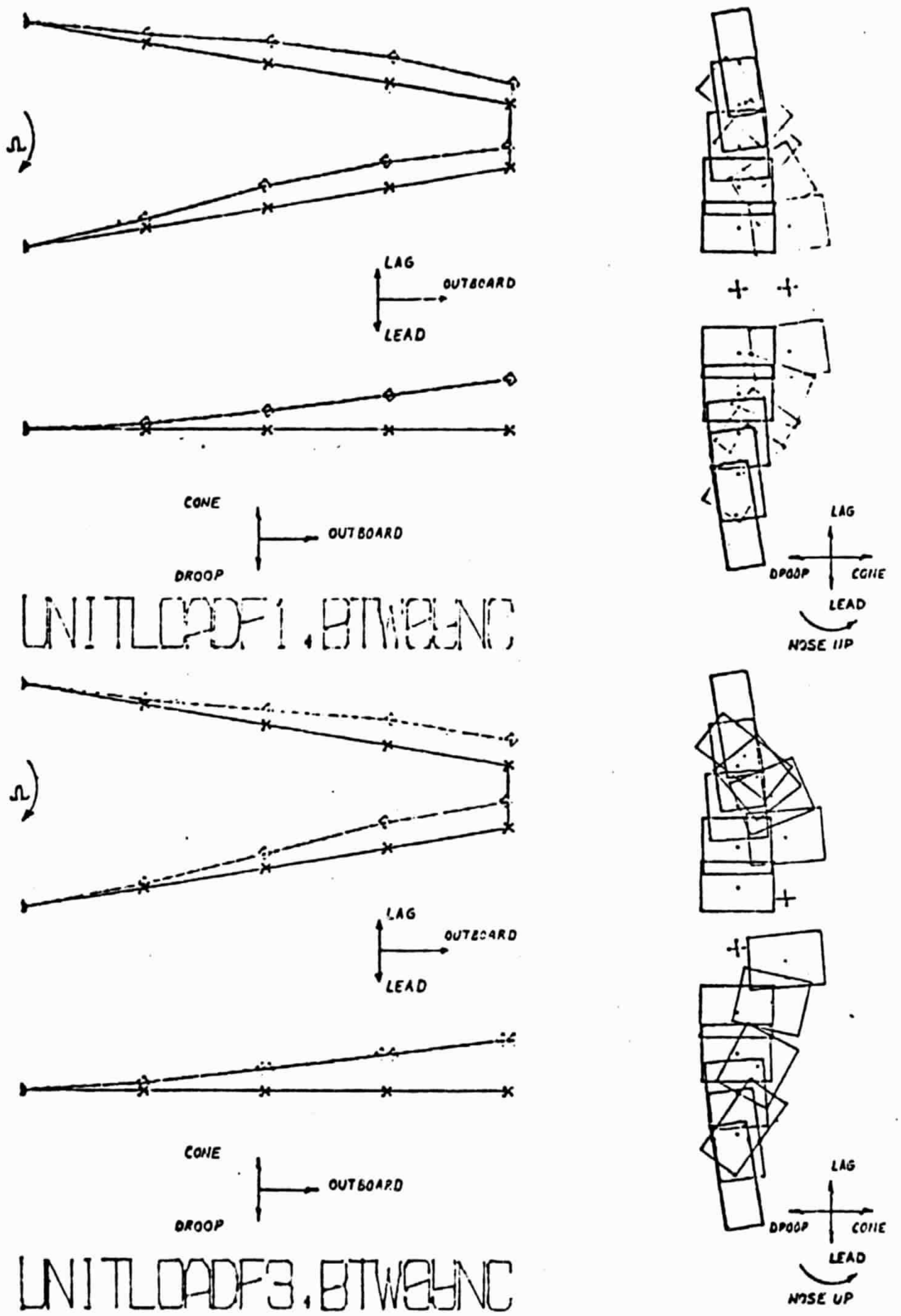



Figure 33. Synchronous Base Twist - Normalized Nodal Deflections.

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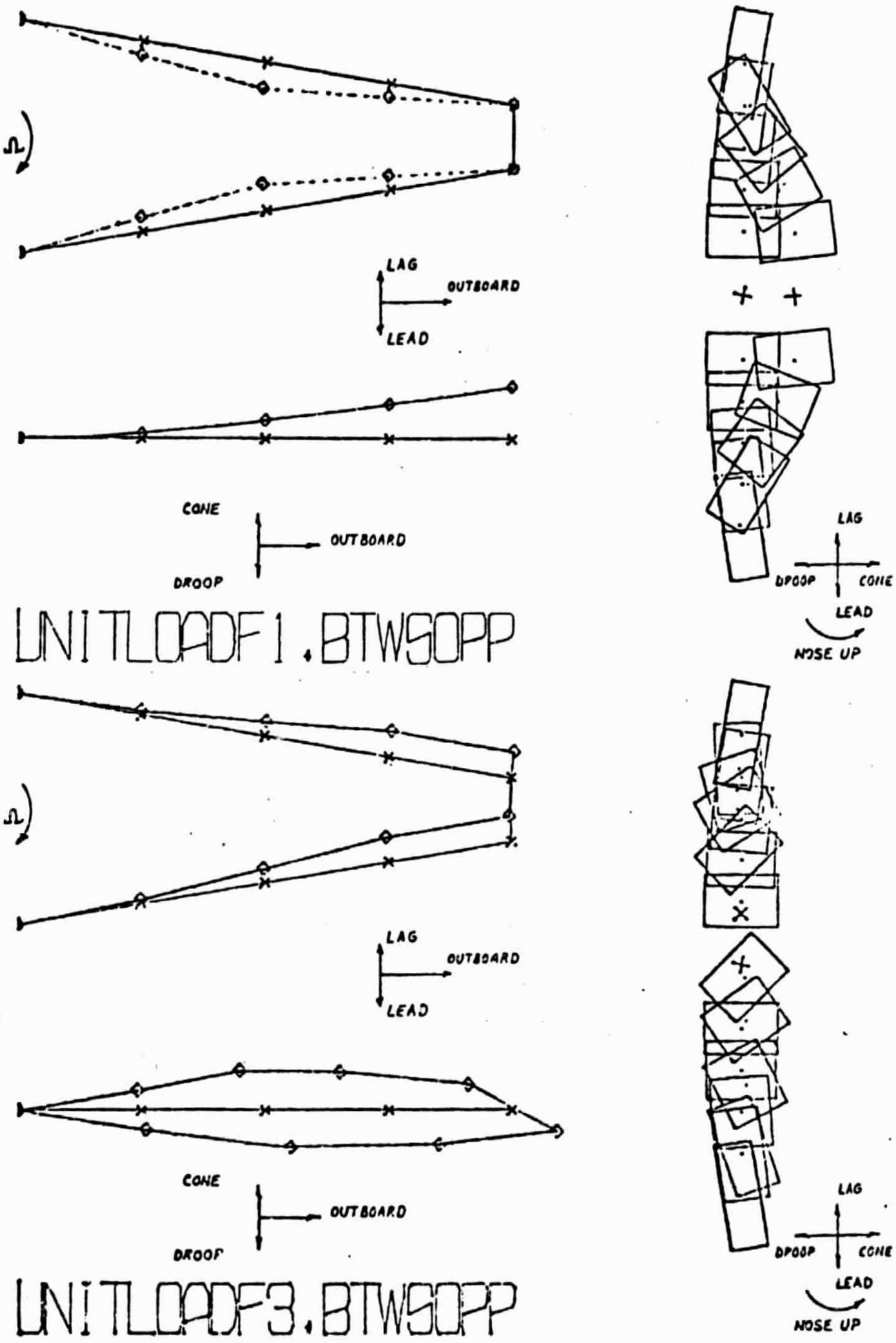


Figure 34. Opposable Base Twist - Normalized Nodal Deflections.

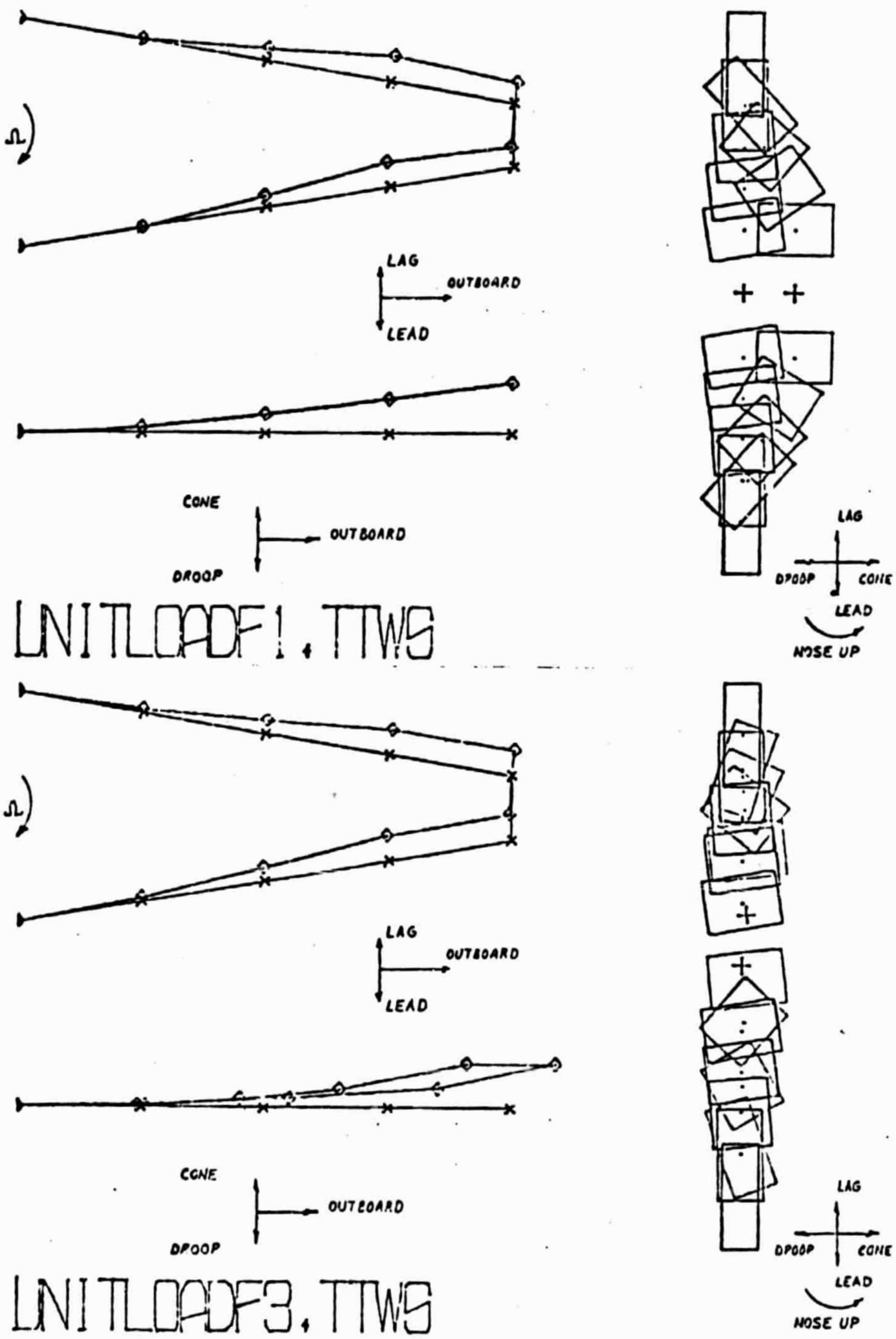


Figure 35. Tip Twist - Normalized Nodal Deflections.

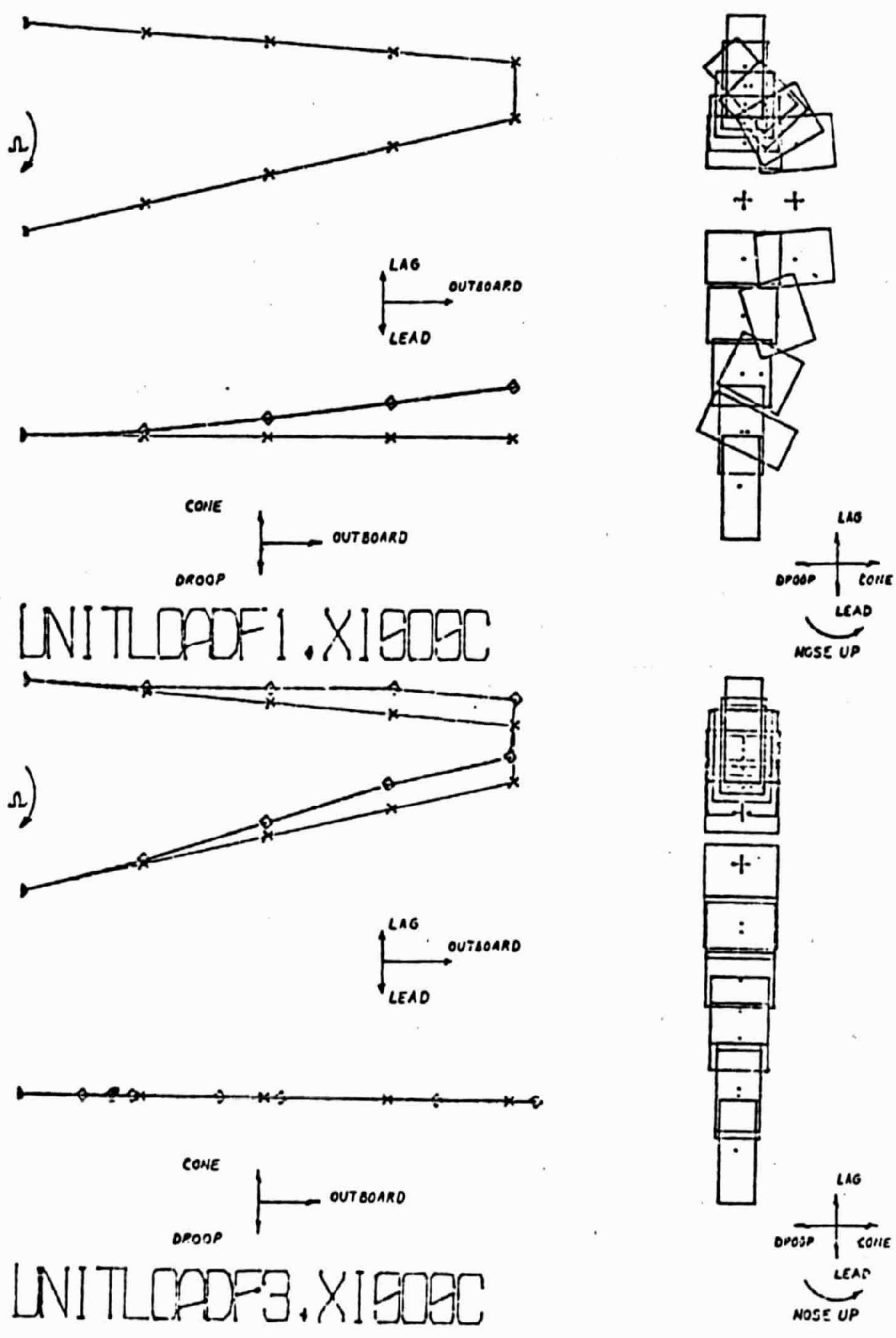


Figure 36. Non-Isosceles Planform - Normalized Nodal Deflections.

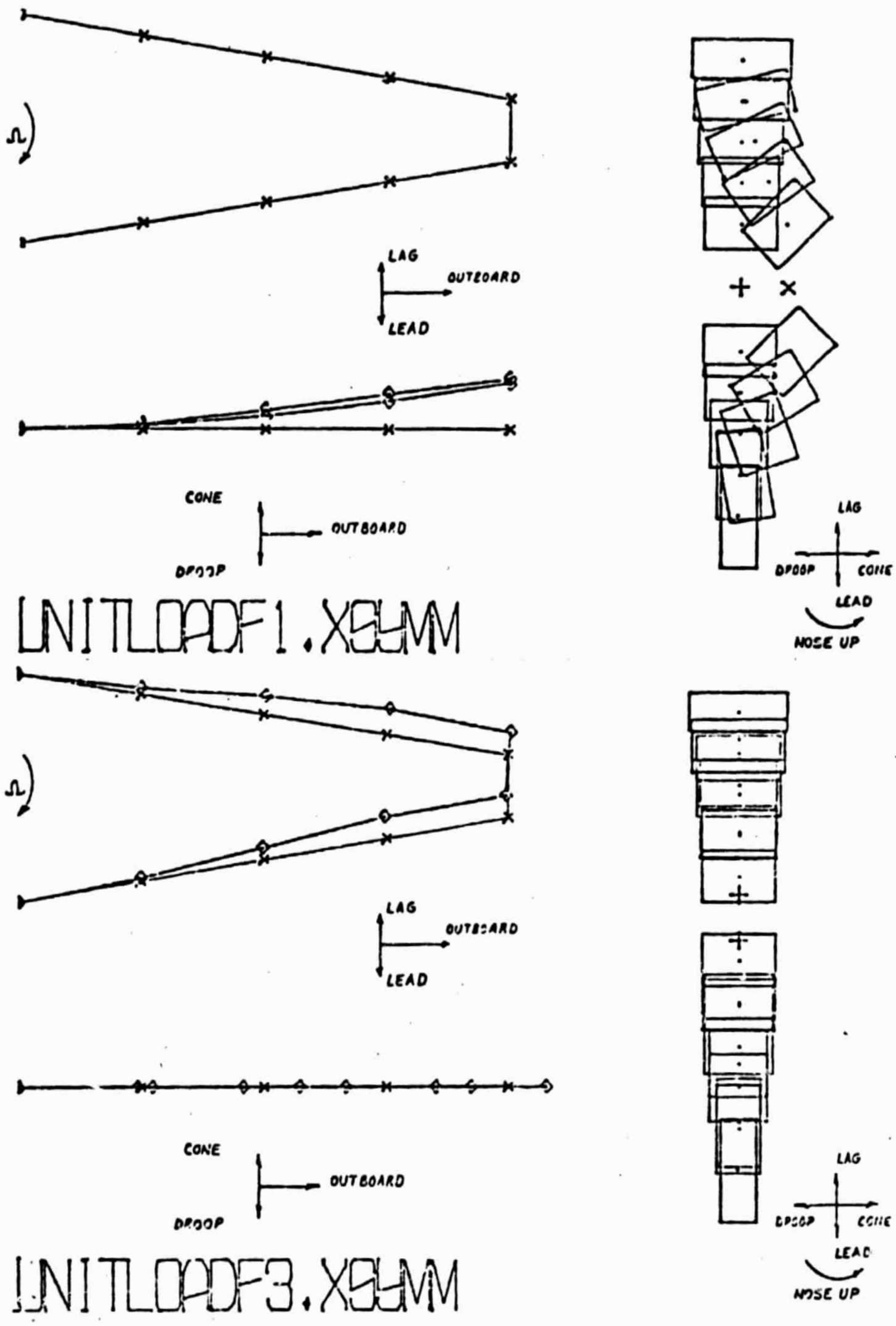


Figure 37. Non-Symmetric Tailoring - Normalized Nodal Deflections.

same scale as flapping deflections. The nodal rotations of the cross-sections are normalized to 45°. These normalizations were chosen because the purpose of the plots was to show kinematic trends in the coupling. If a different goal is in mind, the graphics routine has the option of forcing the same normalization factor for all three views with a scale drawn for measuring "real inches" from the plot. Also, the user may instruct the code to suppress all normalization whence the deflections and rotations are plotted to the same scale as the rest of the drawing.

5.1.8 Alternate Materials. It is useful to briefly outline the advantages and disadvantages of Kevlar and S-glass as opposed to graphite.

(1) Both Kevlar and S-glass have higher tensile ultimate strengths ($F_{tu}$) than graphite.
(2) Kevlar has a lower specific weight than graphite, while S-glass is higher.
(3) Both are softer in bending than graphite.
(4) Kevlar has a better E/G (36.67), while S-glass does not (10.95) (Graphite E/G = 33.62).
(5) Kevlar has a very poor $F_{cu}/F_{tu}$ (0.2), while S-glass (0.58) is more nearly comparable to graphite (0.67). ($F_{cu}$ = compressive ultimate strength).
(6) Both Kevlar and s-glass are less brittle and more tolerant of shock impacts than graphite.
(7) Galvanic corrosion must be considered when graphite touches metal (although it is not expected to in the EPB design).

Because of the great disparity in tensile and compressive strengths of these materials, separate cases were done for tension and compression. The material properties used are listed in Table 7. The procedure used in comparison gave special attention to being totally unbiased toward any particular material. Results of comparative tests can often be inadvertently swayed in a preferred direction by biased choice of criteria.

The graphite baseline critical stress (3g droop) was found as a percentage of ultimate strength. Assuming that the aspect ratios of the cross-sections

TABLE 7. PROPERTIES OF UNIAXIAL FIBER (0°) COMPOSITE

| | S-GLASS (Source) | GRAPHITE (Source) | KEVLAR-49 (Source) |
|---|---|---|---|
| Tensile Ult. ($F_{tu}$) | 200 ksi (Kaman Tests) | 195 ksi (Kaman Tests) | 200 ksi (Dupont) |
| Compressive Ult. ($F_{cu}$) | 115 ksi (Kaman Tests) | 130 ksi (Cyanamid) | 40 ksi (Dupont) |
| Bending Modulus (E) | 7.17 Msi (Kaman Tests) | 19.5 Msi (Kaman Tests) | 11 Msi (Dupont) |
| Shear Modulus (G) | 0.655 Msi (Kaman Tests) | 0.58 Msi (Kaman Tests) | 0.3 Msi (Dupont) |
| Specific Weight ($\gamma$) | 0.07 $lbs/in^3$ (The Literature) | 0.055 $lbs/in^3$ (The Literature) | 0.050 $lbs/in^3$ (Dupont) |

shall be the same as the baseline, the dimensions of the new material root section are determined to keep critical stress at the same percentage of ultimate. Outboard planform and thickness are then scaled from this dimension to the baseline shape. The rigid blade tip deflection was then computed using the hand calculation technique detailed above in Section 5.1.3. A tip deflection less than that of the baseline implies that stress is the critical condition and the sizing for that case is complete. A tip deflection greater than the baseline implies that deflection is the critical condition and the new material must be re-sized to equilibrate the tip deflections. Due to the complexity of the governing equation, this is an iterative process.

The procedure was done for the CEPB. Numerical results for the PEPB would be different, but not the conclusions to be drawn therefrom. If material X were a clear winner for CEPB, it would also be a clear winner for PEPB. The results are compiled in Table 8. Notice that all cases are heavier and consume more drag area than graphite.

Each of the cases defined by this sizing was run through the finite element program to determine the effect on torsional stiffness. As is seen in Figure 38, all of the cases are stiffer than the baseline. Torsional softness, as one criterion, therefore speaks in favor of graphite.

5.1.9 Analytical Conclusions And Recommendations. The elastic pitch beam concept with the modifications described here is a very promising design which circumvents some of the major problems of previous experimental designs. The recommended configuration based on the results obtained in this study is the CEPB with an elastomeric snubber at the inboard end of the blade extension/ torque tube. The snubber should provide radial free slip for the blade, angular misalignment capability able to cope with blade excursions, and changeable (FRR) damping rate characteristics in the lead-lag direction.

A four- or five-ply laminated elastomer assembly should comprise the legs of the EPB. The fixture in the torque transmission area must allow thru slip of the upper and lower plies for cyclic soft mode operation to ensue. These

TABLE 8. EPB PROPERTIES (ALTERNATE MATERIALS)

| | GRAPHITE BASELINE | KEVLAR TENSILE | KEVLAR COMPRESSIVE | S-GLASS TENSILE | S-GLASS COMPRESSIVE |
|---|---|---|---|---|---|
| Critical Condition | N/A | Deflection | Stress | Deflection | Deflection |
| Root Dimensions (Inches) | 4 x 1.5 | 4.61 x 1.73 | 5.92 x 2.22 | 5.21 x 1.955 | 5.12 x 1.925 |
| Tip Dimensions (Inches) | 2 x 3 | 2.31 x 3.46 | 2.96 x 4.44 | 2.61 x 3.91 | 2.57 x 3.85 |
| Running Weight Of One Leg (Lbs/Inch) | 0.330 | 0.399 | 0.657 | 0.734 | 0.711 |
| Critical Stress (Percentage Ultimate) | 27.95% T<br>41.92% C | 17.77% | 41.92% | 12.32% | 22.44% |
| Rigid Tip Deflection (Percentage Baseline) | N/A | 100% | 36.94% | 100% | 100% |



C-2

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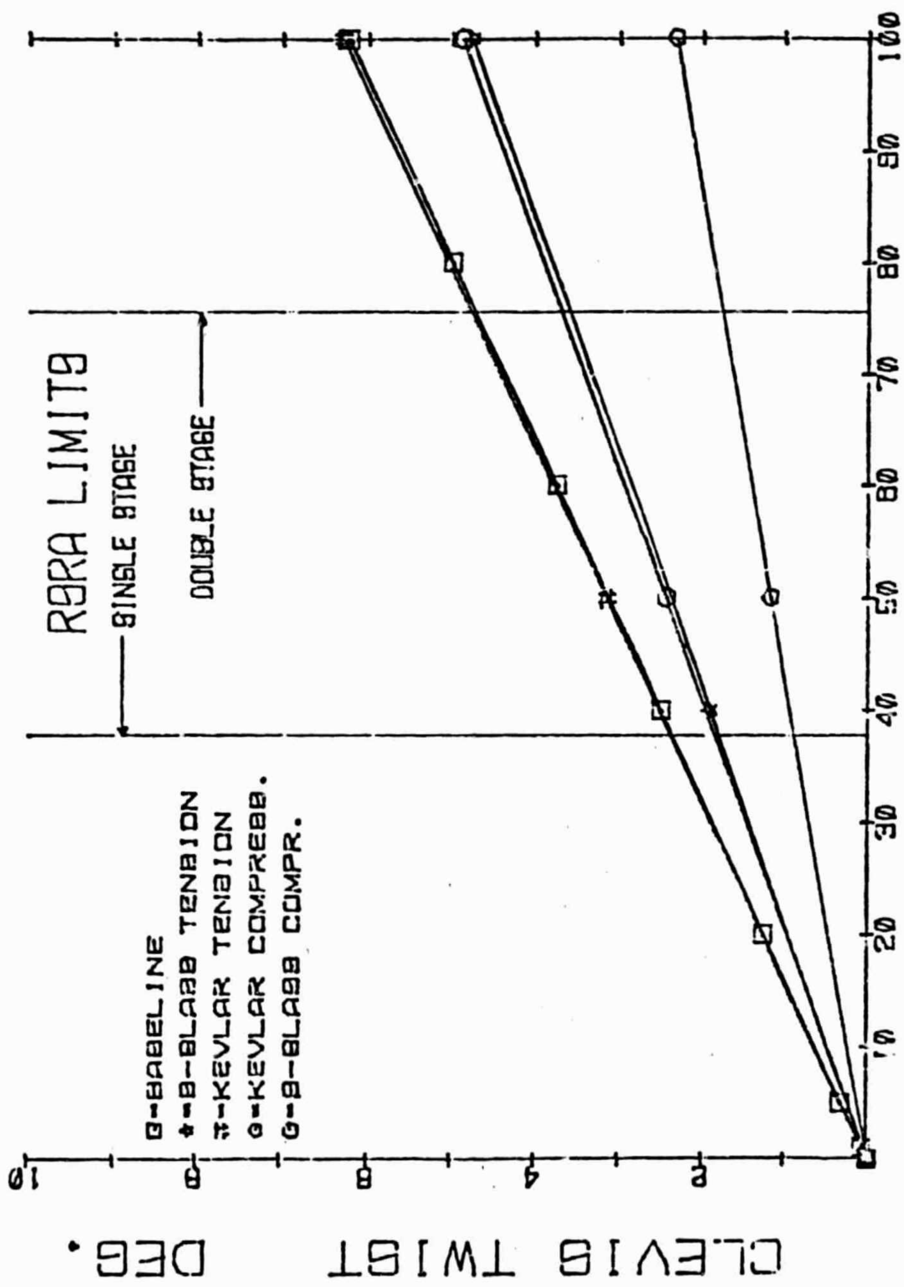


Figure 38. Materials Effect On Torsional Stiffness.

fixtures should also incorporate curved shoes with elastomeric surfaces which continuously adjust the flapwise "leaf spring" stiffness of the EPB legs by shifting the bending hinge region outboard as deflection increases.

Built-in synchronous base twist appears, at this time, to be the most desirable structural coupling technique. It supplies the intended stabilizing couplings - flap-up and nose-up pitch from an input of lag deflection. However, if control or pitching moment is input, pure twisting of the structure results. No flap or lag is coupled with it.

From another point-of-view, built-in opposable base twist is also attractive. Input of in-plane forces produces very strong pitch response but no flap response. Since flap and pitch are de-coupled, the direction is not important to note. For example, lag/nose-down response is changed to lag/nose-up response by simply reversing the angle of opposition. Flap input produces no lag or pitch response, while pitch input produces a weak lag response. The question of which type of base twist would ultimately be chosen must be left to preliminary design.

On the basis of brittleness considerations and impact forgiveness, it is recommended that the top half plies of the EPB be wound from Kevlar. The bottom and center plies should be wound from graphite for compressive integrity in droop. The entire assembly can be coated with urethane for further sacrificial protection and ultraviolet isolation. No advantage could be seen in incorporating S-glass into the design. In fact, fiberglass/graphite make a poor marriage in hybrid composites because of difference in thermal expansion coefficients. Kevlar, however, mates well with graphite in this respect.

Perimetral wraps (belly bands) of Kevlar should be supplied around the rectangular leg sections at one or two midspan locations. This is to prevent local buckling of the thin underleaves during droop. These hoops need not (and should not) be "beefy." Thickness of only a few strands will do.

A final useful addition might be a thin, very flexurally soft Kevlar "underply" beneath the lowest graphite ply. This would not be bonded to the elas-

tomer and would not act unless the graphite were shattered and lost, e.g., a ballistic strike. Since the main Kevlar upper plies alone are more than sufficient for CF retention, the rope-like underply would come into play to balance the offset load caused by the graphite loss (see Figure 39). The affected blade would thus be retained long enough to permit a noncatastrophic landing, although the EPB would possibly buckle and be destroyed once CF was lost.

## 5.2 Design Summary And Observations

5.2.1 Rotor Hub Flat Plate Drag Area. The target for hub drag area, as specified in the SOW, is 2.8 $ft.^2$ for a 16000-lb. aircraft. For the RSRA, with a gross weight of 18,400 lb., the scaled-up target value becomes 3.03 $ft.^2$. Methods cited in Reference 24 were used to calculate the flat plate drag area of the EPB rotor head. The resulting area is 2.43 $ft^2$, which represents a 20 percent reduction from the target of 3.03 $ft.^2$.

Reference 24 cites empirical data for various unfaired hubs showing that the primary trending parameter in determining drag is projected frontal area. The data is trend fitted with a quadratic in terms of $A_p$, projected area. The effective drag coefficient is then approximated by

$$C_{D_H} = .582 + .0349\ A_p - .00057\ A_p^{\ 2}.$$

The area used in computing $A_p$ takes two opposing blades at flat pitch and zero fuselage pitch. Computation is then taken from the end of the grip unit (blade attachment) through the hub center to the opposite grip unit. Conservative allowance was made for any protruding hardware, and pitch horns were included. Exposed rotor shaft length, swash plate, actuators, and pitch links were not considered as these items were assumed to be common to all the configurations investigated.

5.2.2 Parts Count. The target specified for the number of rotor head parts is 50 without consideration for standard fasteners. The SOW is unclear as to whether or not bonded assemblies are to be treated as a single part. Also,

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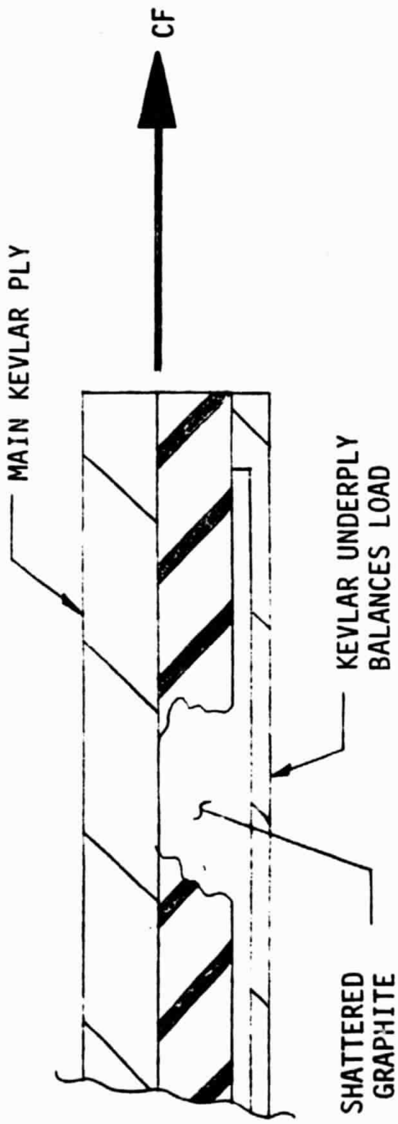


Figure 39. Graphite Loss Causing Offset CF Load.

parts such as the blade attachment bolts are standard in that they are purchased off-the-shelf. They can, however, be considered special as opposed to conventional hardware that can be purchased in any aircraft supply outlet.

Table 9 shows a listing of the rotor head parts with four methods of treating the parts count to account for the two questionable areas cited above. The decision was made to treat bonded assemblies as a single part and to include the blade attachment bolts as special hardware (column 2). The resulting parts count is then 29, representing a reduction of 42 percent from the target.

5.2.3 Weight Estimate. The weight estimate includes all parts of the rotor head (including fasteners) to the point of attachment of the blade, rotor shaft, and pitch links. A summary of the components, materials, unit weights, quantities and total weight is given in Table 10. Factors that could change the weight moderately in final design include geometry and sizing refinement and material changes. It is expected, for example, that the final EPB will be a hybrid of materials, such as graphite and Kevlar. Some metal parts could change based on final stress analysis.

5.2.4 Fabrication Features. The EPB rotor head has many attractive functional features which have been discussed in previous sections. The design philosophy which yielded the developed concept of the EPB has also concentrated on areas important to high volume production and quality control with subsequent benefits in minimum rotor track and vibration problems and inherent reliability.

5.2.4.1 Pitch Beams - The pitch beam is the central member of the rotor head, and because it is a composite construction it can be subject to either nonuniformity or near-exact reproduction, dependent on simplicity in design, tooling control, and quality control. The use of fibers in the design is straightforward and uses well-established techniques that can be translated easily into predictable tooling. Existing resin systems should be completely satisfactory for the application. The use of elastomer is restricted to nonstructural application in a manner than should ensure integrity of bonds.

TABLE 9. PARTS COUNT

| COMPONENT | #1 QTY REQ. | #2 QTY REQ. | #3 QTY REQ. | #4 QTY REQ. |
|---|---|---|---|---|
| ELASTIC PITCH BEAM | 2 | | 2 | |
| CLEVIS | 4 | 2 ASSYS. | 4 | 2 ASSYS. |
| TORQUE TUBE | 4 | | 4 | |
| PITCH HORN | 4 | 4 | 4 | 4 |
| SHOE - PLAIN | 2 | 2 | 2 | 2 |
| SHOE - PIVOT | 8 | 8 | 8 | 8 |
| HUB | 1 | 1 | 1 | 1 |
| BEARINGS-CONTROL PIVOT | 4 | 4 | 4 | 4 |
| BOLTS-BLADE ATTACH. | 8 | 8 | - | - |
| TOTALS | 37 | 29 | 29 | 21 |

CONTRACTORS UNDERSTANDING OF GROUND RULES INDICATES COLUMN #4 IS CORRECT, HOWEVER, COLUMN #2 IS CONSIDERED MORE REASONABLE. BLADE ATTACH BOLTS ARE CONSIDERED SPECIAL EVEN THOUGH OFF-THE-SHELF.

TABLE 10. EPB PRELIMINARY WEIGHTS

| COMPONENT | MATERIAL | UNIT WEIGHT-LBS | QUANTITY REQUIRED | TOTAL WEIGHT-LBS |
|---|---|---|---|---|
| ELASTIC PITCH BEAM | GRAPHITE/EPOXY | 71.5 | 2 | 143.0 |
| CLEVIS | 7075 ALUMINUM | 20.6 | 4 | 82.4 |
| TORQUE TUBE | GRAPHITE/EPOXY | 17.0 | 4 | 68.0 |
| PITCH HORN | 7075 ALUMINUM | 6.7 | 4 | 26.8 |
| SHOE-PLAIN | 7075 ALUMINUM | 16.8 | 2 | 33.6 |
| SHOE-PIVOT | TITANIUM | 12.5 | 4 | 50.0 |
| HUB | TITANIUM | 29.0 | 1 | 29.0 |
| BEARINGS-CONTROL PIVOT | STEEL/TEFLON | .45 | 4 | 1.8 |
| BOLTS & NUTS - BLADE RETENTION | STEEL-1.25 DIA. | 1.93 | 8 | 15.4 |
| BOLTS & NUTS - 1. HUB/SHOE | STEEL - 5/8 DIA. | .34 | 16 | 5.4 |

TOTAL ROTOR HEAD WEIGHT = 455.4 LB.

Development of new techniques in bonding should not be necessary, and the design is tolerant of elastomer degradation. With adequate quality specifications, detailing acceptance criteria for all materials, accompanied by rigid guidelines for material preparation and application, minimum problems are anticipated in fabrication.

5.2.4.2 Torque Tube - Although the torque tube is wound in a different manner than the pitch beam, the technique is well proven for the type of component. The same considerations cited above for tooling and material control apply for reproducible parts. Final bonding of the pitch beam, torque tube, and clevises will require closely controlled tooling.

5.2.4.3 Attachment Components - All components such as clevises, shoes, pitch horns, and the hub are designed to be simple shapes, amenable to numerically controlled machining methods with minimum waste. Use of expensive forgings has been avoided.

5.2.5 Materials - Cost Factors. Materials that were used in the conceptual design that could be considered high cost are graphite and titanium. With the brief time period allowed, it was necessary to use materials that were reasonably certain to answer most of the goals without optimization.

5.2.5.1 Graphite - Graphite was chosen early in the study for both pitch beams and torque tubes because of its structural properties which lead to minimum size and weight. There is a definite, large cost penalty for graphite when compared to materials such as Kevlar and S-Glass. A limited effort was spent at the conclusion of the study to assess the feasibility of using these other materials which resulted in some interesting observations, summarized below.

The use of either Kevlar or S-glass as the sole material for the pitch beam causes the structure to grow both in size and weight to unreasonable proportions. Kevlar also has very poor compressive capability which impacts the static droop problem. It is possible, however, that Kevlar could be used in

the laminations in the upper beam section and substantially reduce the amount of graphite needed to maintain overall structural requirements. A small growth in size and weight would result, but a detailed trade-off analysis might show the cost benefit to outweigh the disadvantages.

A second benefit to the use of Kevlar is its impact resistance, which is high compared to graphite. The poor impact characteristics of graphite would require some protective covering for normal maintenance protection even if a hybrid of graphite and Kevlar is not finally chosen. A urethane spray with high solids content would probably suffice for maintenance and environmental protection.

Another interesting possibility, for a mix of a thin laminate of Kevlar on the lower surface with laminates of graphite followed by upper laminates of Kevlar, is the potential improvement in vulnerability characteristics. If the graphite acted as a sacrificial layer to expend the energy of a "hit", the limited analysis performed indicates that the Kevlar might retain the unit intact for a period long enough to land. The low compressive strength of the Kevlar would, however, very likely cause buckling when C.F. is lost at shutdown. The potential benefits resulting from a hybrid composite justify a thorough trade-off study during the preliminary design phase.

5.2.5.2 Titanium - Titanium was used for highly loaded metal components such as the hub and the pivot-shoes for weight savings. Also, titanium is a good choice for contact with graphite due to galvanic corrosion considerations. Aluminum must be completely isolated. Again, a trade-off study will be necessary in the preliminary design phase to determine whether or not the weight savings justify the cost penalty. Another consideration in a production application is that the hub of the conceptual design is more complex because of the need to adapt to the RSRA hub. In production, the mating surfaces of both rotor shaft and rotor head attachment would be designed for minimum cost.

5.2.6 Special Hardware - Cost Factors. Two items of hardware that could be termed "special" are anticipated in the EPB design: the quick-release bolt that attaches the blade to the clevis and the bearing at the inboard end of the torque tube.

5.2.6.1 Blade Attachment Bolt - These bolts have special features that make them extremely attractive from both maintenance and operational standpoints. The blade attachment point, for operational considerations, must be an extremely tight fit that provides bending continuity across the joint. Any looseness invites vibration and wear. With conventional hardware, this requirement dictates extremely close tolerances and presents a problem whenever the blade is either folded or removed.

A special bolt with an expandable bushing has been used successfully in the Blackhawk, the Hughes 500, and other modern aircraft. This bolt has a quick-release feature with a positive lock when engaged. The benefits are obvious: blade installation, removal, and folding time are minimal, and close tolerances for fabrication are eased. The additional cost of the bolt is saved many times over in both manufacture and field use.

5.2.6.2 Torque Tube Pivot Bearing - The pivot bearing is actually a combination of a spherical and a sliding bearing. The spherical bearing is necessary for freedom in flapping motions, while the sliding feature is necessary to permit axial motion. Nonlubricating Teflon bearings have been used in previous hingeless design, but wear characteristics have resulted in short lives. A nonlubricating bearing having both spherical and sliding features was used in the DAVI Rotor Isolation Program (Reference 25) under a much more severe vibratory loading environment; virtually no wear was indicated after a fatigue test of $5 \times 10^6$ cycles on six bearings. This type of bearing will be the primary candidate for the EPB. Elastomerics are also a possibility, but the required angular deflection may prohibit use.

5.2.7 Environmental Factors. The EPB has excellent characteristics for environmental factors. A coat of impact-resistant urethane will protect the

structure against sand, dust, rain, hail, and ultraviolet effects. Careful selection of resins, bonding agents, and elastomers will protect against high and low temperature extremes. Ice should be easily shed with the flexing of the composite members. Conductive paths for lightning protection can easily be provided without the problem of bridging large rotating bearings.

5.2.8 Maintainability Features. Maintenance actions for the EPB include installation, removal, inspection, blade folding, and replacement of the pivot bearing damper assembly. Blade installation and removal is a simple task requiring a blade sling and the installation or removal of two quick-release bolts. Blade folding is accomplished, as on the Blackhawk or Starflex, by means of a special support tool which attaches to the center of the rotor head and then is secured at each blade attachment point by removing the quick-release bolt and inserting a pivot pin. The other quick-release bolt is then removed and the blade is rotated on the pivot pin while supporting the blade on a conventional support crutch.

With blades removed, the rotor head can easily be removed or installed as a single unit. It can also be removed or installed by separating or joining its few component parts.

The only part requiring periodic replacement is the pivot bearing/damper assembly. This part is expected to have a very long life, but projections cannot be made without preliminary design and analysis effort. It is expected that all four assemblies could be replaced in less than one hour.

Inspection involves only the normal visual preflight and postflight requirements for rotor system components.

5.2.9 Reliability/Long Life Features. Long life is generally a characteristic of composite structures given due consideration in design for loading conditions and use of proven techniques in the fiber orientation and experience with the particular resin system. In the case of the EPB, as in other composite flex beams, weaknesses recognized include possible damage from

environmental factors and dictate the use of the protective urethane covering. Particular attention also has to be paid to any surfaces subject to fretting, such as the stress distributing shoes, which have an elastomer layer for protection. All faying surfaces of composite and metal are given this protection.

Attachment points are minimized throughout the system. In addition, the design is such that tolerances can be relaxed and replacement of usually high usage fasteners minimized.

The only component in the system with an expected life shorter than the 10,000-hour life of the pitch beam is the pivot bearing/damper assembly, and this part is expected to have a very long life by bearing standards.

5.2.10 FRR Variations. Rotor head variations that are low in cost and easily flight qualified are somewhat limited. Changes to the damper assembly for lead-lag damping and to control stiffness are reasonable to expect and will probably be desirable for both ITR and FRR. Major changes could include a complete change of beam assemblies. If this type of change is contemplated, a wide variety of changes in laminates and elastomer layers is possible. Each change, however, would probably require almost complete requalification and would be very costly.

## 6.0 DYNAMIC CONSIDERATIONS (ROTOR HEAD RELATED ONLY)

### 6.1 Introduction

There has been a considerable effort on the part of both government and industry over the past years to develop workable solutions to hingeless rotor design. Considering the extensive goals of the ITR program to combine the latest state-of-the-art concepts into one design, a parallel company-supported effort was initiated to enhance the work performed in the conceptual definition study. One task of this effort was to review all available literature on the subject. Most of the findings were supportive of each other differing only in the degree of importance attached to certain stabilizing effects. The

following paragraphs summarize the findings of the literature search. The results of the search are being used to aid planning of the preliminary design phase.

## 6.2 Literature Search - Summary Of Findings

Articulated rotors are characterized by hinges and bearings which attach the blades to the hub and dampers which are used to prevent dynamic instability. These components are subjected to high steady and vibratory loads, and, as a result, require frequent inspection, maintenance, and replacement. To improve this situation, hingeless rotors are being designed to reduce mechanical complexity and thereby improve maintainability and reliability. Additionally, the hingeless rotor generally improves helicopter stability characteristics by providing an increase in control power and angular rate damping compared to articulated systems.

The advent of composite materials has accelerated the development of hingeless rotors. Composite designs hold the promise of tailoring retention system properties to achieve favorable aeroelastic couplings with resulting improvements in stability over the full flight spectrum.

One type of instability which can be encountered by articulated rotor helicopters is ground resonance. When the frequency of the rotor fundamental lead-lag regressing mode is in proximity to an airframe pitch or roll mode frequency, on the ground, the potential for the phenomenon exists. Generally, incorporation of blade in-plane dampers or modification of landing gear stiffness and damping characteristics ensures stability. Hingeless rotors, characterized as soft in-plane ($.6\Omega$ - $.8\Omega$), are susceptible to air resonance instability as well as ground resonance instability. This phenomenon occurs when the rotor fundamental lead-lag regressing mode frequency coalesces with a fuselage pitch or roll frequency in the air.

Hingeless rotor air resonance instability can be avoided by introducing damping into the system. Aeroelastic couplings can be designed into the

system to introduce aerodynamic damping, reducing the potential for instability. Additionally, rotor structural damping may be increased by using elastomeric damping material.

A great deal of research has been conducted to aid in understanding air and ground resonance instability characteristics of hingeless rotors. Theoretical developments have yielded mathematical models and analytical techniques capable of predicting hingeless rotor stability response. Experimental research has provided insight into design variables, including structural couplings, in addition to providing a data base for use in correlation of analytical simulations of the problem.

Analytical and experimental work indicates that there are several design parameters that can be used to influence air and ground resonance instability. These include sweep, precone/droop, control system stiffness, blade in-plane and out-of-plane fundamental frequencies, kinematic and structural couplings, and elastomeric damping. Additionally, the influence of operating parameters such as rotational speed, collective pitch, airspeed, and rotor shaft angle has been studied.

There has generally been agreement that ground and air resonance instability would be suppressed if a system displayed sufficient rotor in-plane and/or body damping. Use of external dampers has been discouraged, as this defeats the ultimate purpose of hingeless systems. Use of elastomers for damping in sophisticated structures had not developed to the point of reasonable risk until the late 1970's; effort prior to that period had concentrated solely on the use of aeroelastic couplings. Recently, effort has concentrated on a combination of elastomeric damping and aeroelastic coupling.

Elastomeric damping techniques were applied effectively in the work reported in Reference 4 where installation of elastomer damping increased stability in hover and at 90 knots. However, Reference 16 indicates that the effects of structural damping on ground resonance "in vacuo" are influenced by the ratio of blade in-plane damping to body damping. Further experimental programs are

required and are planned to establish the beneficial effects of elastomeric damping.

Generally, hingeless rotor systems have lacked sufficient structural damping to preclude air and ground resonance instability. Additionally, at low thrust levels the aerodynamic damping contribution is small, requiring even more careful treatment of aeroelastic coupling effects in design. Couplings considered include a combination of kinematic pitch-lag and pitch-flap coupling and structural flap-lag coupling. Reference 20 presents fundamental theory regarding aeroelastic couplings, concluding that stability of isolated rotor systems at low thrust levels may be improved by combining elastic flap-lag coupling and pitch-lag coupling. The combination of these couplings is important, since there is insignificant or no increase in lead-lag damping when the couplings are applied individually. The importance of the combination of pitch-lag and flap-lag coupling is verified in Reference 16. Experimental data of Reference 6, which are an approximate equivalent of an air resonance condition, indicate that addition of negative pitch-lag coupling increases the lead-lag regressing mode damping for a rotor where the flapping stiffness is less than the lead-lag stiffness. However, subsequent addition of structural flap-lag coupling did not produce a significant damping improvement.

Precone, sweep, and control system stiffness can be used to introduce damping. Reference 11 indicates that precone affects pitch lag coupling by providing in-plane damping and increasing air resonance stability. Experimental and analytical results in Reference 13 concur with the findings of Reference 11. Regarding blade sweep, Reference 8 shows that aft sweep does not increase damping at low blade angles, whereas at higher blade angles aft sweep can reduce damping. This effect occurs in hover and at 60 knots.

Control system stiffness also affects lead-lag regressing mode damping. Reference 22 considers the effect of control system stiffness in conjunction with precone on an isolated rotor in hover. At low thrust levels and zero precone, reduction in control system stiffness is destabilizing. Conversely, as thrust level increases, the reduced stiffness control system displays an increase in

damping with accompanying improvement in stability.

As indicated in previous sections, the aeroelastic behavior of hingeless rotors is dependent on the proper use of many parameters, including elastic coupling through blade precone and sweep, control system stiffness, and structural damping through use of elastomer. A parametric study is necessary to determine the impact of combinations of the primary parameters on stability in all operating conditions; in addition, it is necessary to evaluate the impact of all candidate parameters on performance, vibration, loads, and control forces when combined with blade geometrical and structural parameters.

## 7.0 SPECIFICATION REQUIREMENTS COMPARISON

### 7.1 Vulnerability

Adequate assessment of vulnerability requires a more extensive effort than is permitted by the scope of the present study. A merit factor of 1 has been used in calculating the merit function. A full vulnerability evaluation will be required in the preliminary design phase.

### 7.2 Risk Of Aeromechanical Instability

Aeromechanical instability will be dependent on a concepts capability to furnish adequate damping resulting from elastomeric damping material and/or the necessary combination of elastic coupling. The prototype configuration of the EPB rotor head will have the proper mix of these controlling factors as determined by analysis, specimen test, and scale model test. In addition, the configuration will have a wide range of variability for the factors contributing to damping which should insure adequate stability in the final government flight-tested configuration. A merit factor of 1 has therefore been designated.

### 7.3 Hub Drag Area

The hub drag area was established in Section 5.2.1 as 2.43 $ft^2$. The scaled-up target figure for an 18,400-lb gross weight vehicle is 3.03 $ft^2$. The reduction from the target figure is therefore 20 percent, giving a number of 20 for the merit factor.

### 7.4 Hub Weight

The scaled-up weight target for an 18,400-lb vehicle is 460 lbs. The weight estimated for the EPB is 455.4 lbs. This represents a reduction of 1 percent from the target or a figure of 1 for the merit factor.

### 7.5 Parts Count

The number of parts established for the EPB is 29, resulting in a 42 percent reduction from the target of 50 (without standard fasteners). The merit factor is therefore 42.

### 7.6 Rotor Hub Moment Stiffness

The scaled-up hub moment stiffness goal is 172,500 ft-lb/rad for an 18,400-lb vehicle starting with a goal of 150,000 ft-lb/rad for a 16,000-lb aircraft. The use of elastomer laminations in the configuration, envisioned in the EPB, permits achieving the goal with a reasonable degree of accuracy. Since the merit factor is based on achieving the goal within an accuracy of $\pm$ 20 percent, it is anticipated that the goal can be met and a figure of 5 is assigned to the factor.

### 7.7 Minimum Rotor Hub Moment, Minimum Rotor Hub Tilt Angle, Fatigue Life

The goals for these parameters are related to rotor head life. Detailed stress analysis was specifically excluded from the concept definition study; however, design for all composite structures in the rotating helicopter system

in the recent past has used similar goals for a 10,000-hour life. Achievement of these goals has been successfully demonstrated through qualification test on production components. The simplified calculations performed in the concept definition phase were based on the same factors used in detail analysis, so it is reasonable to believe that the goals can be met. No credit can be taken, however, for exceeding the goals without detail analysis. The merit factors for minimum hub moment, minimum hub tilt, and fatigue life are therefore 0, 0, and 10, respectively.

### 7.8 Reliability

The goal for reliability of a mean time between removal of 3000 hours is a normal requirement for the state-of-the-art in design practices. A merit factor of 10 has been assigned based on meeting the goal but, again, predictions of exceeding the goal cannot be made without more detailed design and analysis.

### 7.9 Manufacturing Cost

No specific goal was established for cost except that it should be the lowest possible for production without unduly compromising factors affecting life-cycle cost. To establish the merit factor, however, both a target and a baseline had to be determined.

The target was arbitrarily set at $100,000. This was not completely without basis, as it represents better than a 50 percent reduction over modern articulated rotor heads, typical of previous composite design benefit. The baseline was more difficult to determine, and it must be remembered that there was no program allowance for detailed cost estimating necessitating broad ROM (rough order of magnitude) techniques.

The Starflex was chosen as a baseline, state-of-the-art, modern rotor head for which approximate cost and cost/pound numbers were known. Also, the production quantities for Starflex are similar to those anticipated for a

production ITR. The Starflex, however, is a 3-bladed, 4300-lb vehicle compared to the desired 4 bladed, 18,400-lb vehicle. The hub weight for the 4300-lb aircraft was known to be 117 lb. and this was increased by slightly less than a third to account for the 4th blade. This weight turned out to be 3 percent of the increased aircraft gross weight and was converted to cost by using the established cost/pound. The hub weight for the 18,400-lb Starflex equivalent was then determined on the basis of the established 3 percent factor. The resultant weight was converted to cost using the Starflex cost/pound. All figures used in the calculation are shown in Table 11B.

Having established the baseline, cost for the EPB had to be estimated without detailed costing techniques. This was accomplished by grouping components of the Starflex vs. components of the EPB and determining roughly, by fabrication technique and part complexity, whether cost would be higher, lower, or the same for the EPB. This assessment is shown in Table 11A. Based on this assessment, it was judged whether or not the EPB total cost would be higher, lower, or the same and roughly the percentage of reduction or increase.

It was estimated that there would be approximately a 15-percent saving for the EPB, and this percentage was applied to the Starflex cost/pound figure. The hub weight of the EPB was known, and so the rotor head cost was established. Using the evaluation criteria scale, a merit factor of 8.3 was determined.

Obviously, any of the numbers used can be questioned when using such crude techniques. There is a fair degree of confidence, however, based on the evaluation of components and fabrication technique, that the ultimate cost of the EPB would be less than that of a similar size Starflex.

### 7.10 Auxiliary Lead-Lag Damping

The evaluation criteria are based on the ability to add damping if required. This feature is inherent in the final configuration of the EPB, so the highest grade of 2 was selected for the merit factor.

## TABLE 11A. MANUFACTURING COST ASSESSMENT

COST CONSIDERATIONS - 18,400 LB. GROSS WEIGHT VEHICLE - ROTOR HEAD

| STARFLEX (4 BLADES) | | | ELASTIC PITCH BEAM | | |
|---|---|---|---|---|---|
| QTY | COMPONENT(S) | FABRICATION TECHNIQUE | QTY | COMPONENT(S) | FABRICATION TECHNIQUE |
| 1 | STAR | LAYUP WITH SLEEVES AND BUSHINGS | 2 | PITCH BEAMS | WOUND LAMINATES-FINAL BOND (LOWER COST) |
| 6 | SLEEVES | DOUBLE WOUND STRUCTURE | 4 | TORQUE TUBES AND CLEVISES | WOUND STRUCTURE & NC MACHINE (LOWER COST) |
| 4 | PITCH HORN | STEEL STAMP MAY CHANGE WITH SIZE | 4 | PITCH HORN | ALUMINUM-NC MACHINE (SAME COST) |
| 4 | ELAST. DAMPER | VULCANIZED ASSEMBLY | 4 | ELAST. DAMPER | VULCANIZED ASSEMBLY (SAME COST) |
| 4 | SPHERICAL BRG. | --- | 4 | SPHERICAL BRG. | --- (SAME COST) |
| 4 | ELAST. BRG. ASSY. | VULCANIZED ASSEMBLY | | NONE | (NO COST) |
| 4 EA. | SPACERS, BOSSES, BRG. HSGS. | NC MACHINE | 6 | SHOES | NC MACHINE-FEWER PARTS (LOWER COST) |
| 1 | TOP PLATE | NC MACHINE | 1 | HUB | NC MACHINE (HIGHER COST) |

TABLE 11B. MANUFACTURING COST ASSESSMENT

| FACTOR | STARFLEX 3 BLADES | STARFLEX 4 BLADES | STARFLEX 4 BLADES | EPB 4 BLADES |
|---|---|---|---|---|
| GROSS WEIGHT | 4300 LB | 4328 LB. | 18,400 LB. | 18,400 LB. |
| HUB WEIGHT | 117 LB. | 145 LB.(3%) | 552 LB.(3%) | 455 LB.(2.5%) |
| HUB COST | $33,000 | $40,890 | $155,664 | $109,200 |
| COST/LB. | $282 | $282 | $282 | $240 (15% SAVING) |

EVALUATION CRITERIA SCALE 1 - 10

BASELINE (STARFLEX) TO TARGET RANGE $155,664 - $100,000

EPB $109,200 or 8.3 MERIT FACTOR

### 7.11 Torsional Stiffness

A simplified analysis of control force requirements indicates a dramatic reduction in control force as laminations are added to the solid rectangular beam, and the requirement of staying within one and one-half times the normal force requirement should be easily met. On that basis, the merit factor of 0 was selected.

It must be remembered that retrofitting a hingeless rotor head to an existing helicopter using a factor of one and one-half for control force is substantially eroding the normal margin of safety used in design. The stationary controls, hydraulics, and support structure for attachments may have to be redesigned to regain the necessary margin.

### 7.12 Summary

Table 12 presents a summary of goals for both the 16,000-lb vehicle and the 18,400-lb vehicle with the estimated values achieved for the EPB. Differences between the PEPB and the CEPB are relatively minor, and considering the accuracy of the calculations, the merit factors and resulting merit function should apply approximately to both. Merit factors and merit function are shown in Table 13.

## 8.0 CONCLUSIONS

### 8.1 Plain Elastic Pitch Beam (PEPB) Vs. Classic Elastic Pitch Beam (CEPB) -

There is little benefit in retaining the PEPB configuration. Originally it was thought to be a cleaner configuration for drag, lower in weight, and fewer in parts. This has not been the case. More accurate sizing shows that both configurations are practically the same in these areas. In addition, the best design for all interface attachments is the same for each. The major difference is considered a disadvantage for the PEPB in that the unsupported torque tube grows in size and weight and an ideal location for damping material is deleted.

TABLE 12. ROTOR HUB TECHNICAL GOALS - EPB

DESIGN GROSS WEIGHT 18,400 POUNDS

| FACTOR | GOAL (16,000 LB.) | GOAL (18,400 LB.) | ESTIMATED ACTUAL |
|---|---|---|---|
| FLAT PLATE DRAG AREA | 2.8 $FT^2$ | 3.03 $FT^2$ | 2.43 $FT^2$ |
| HUB WEIGHT | 400 LB. | 460 LB. | 455 LB. |
| PARTS COUNT | 50 | 50 | 29 |
| HUB MOMENT STIFFNESS | 150,000 FT.LB/RAD ±20% | 172,500 FT.LB/RAD ±20% | 172,500 FT.LB/RAD ± 20% |
| HUB MOMENT (FAT. CONSID.) | 10,000 FT.LB. | 11,500 FT.LB. | 11,500 FT.LB. |
| HUB TILT ANGLE (FAT. CONSID.) | 5 DEGREES | 5 DEGREES | 5 DEGREES |
| AUX. LEAD-LAG DAMPING PROV. | DESIRABLE | DESIRABLE | PROVISIONS AVAILABLE |
| TORSIONAL STIFFNESS (CONTROL FORCE) | Less than 1.5 x Veh. | Less than 1.5 x Veh. | Less than 1.5 x Veh. |
| FATIGUE LIFE | 10,000 HOURS | 10,000 HOURS | 10,000 HOURS |
| MEAN-TIME-BETWEEN-REMOVAL | 3,000 HOURS | 3,000 HOURS | 3,000 HOURS |
| MANUFACTURING COST | --- | $100,000 | $109,200 |

TABLE 13. MERIT FACTOR/MERIT FUNCTION, ELASTIC PITCH BEAM

| | | | |
|---|---|---|---|
| VULNERABILITY | $K_v$ | = | 1 |
| RISK OF AEROMECHANICAL INSTABILITY | $K_a$ | = | 1 |
| HUB DRAG AREA (%) | $K_d$ | = | 20 |
| HUB WEIGHT (%) | $K_w$ | = | 1 |
| PARTS COUNT (%) | $K_p$ | = | 42 |
| ROTOR HUB MOMENT STIFFNESS | $K_e$ | = | 5 |
| MINIMUM ROTOR HUB MOMENT | $K_m$ | = | 0 |
| MINIMUM ROTOR HUB TILT ANGLE | $K_b$ | = | 0 |
| RELIABILITY | $K_r$ | = | 10 |
| MANUFACTURING COST | $K_c$ | = | 8.3 |
| FATIGUE LIFE | $K_f$ | = | 10 |
| AUXILIARY LEAD-LAG DAMPING | $K_z$ | = | 2 |
| TORSIONAL STIFFNESS | $K_s$ | = | 0 |

$$\text{MERIT FUNCTION} = K_v \times K_a \times (K_d+K_w+K_p+K_e+K_m+K_b+K_r+K_c+K_f+K_zK_s)$$

$$= 1 \times 1 \times (20+1+42+5+0+0+10+8.3+10+2+0)$$

$$= 98.3$$

## 8.2 Probability Of Meeting Specification Goals

The CEPB has the highest probability of meeting the specified goals with a configuration evolving from preliminary design. The PEPB has more unanswered questions at this point in its development.

## 8.3 Stability

Considering the basic features of elastomeric damping and damping from elastic coupling that can be built into both EPB's, combined with the ability to vary each over a wide range, stability should be ensured; although the PEPB will have to rely more heavily on elastic coupling for damping.

## 8.4 Structural Adequacy

Ample for each configuration.

## 8.5 Vulnerability

The EPB basic structure schematized in Figures 18 through 20 (common to both configurations) should be equal to any hingeless rotor head in survivability to a catastrophic hit from 23mm HEI projectiles. Selection of materials for beam laminates in preliminary design combined with ballistic tests may result in improved capability.

## 8.6 Drag, Weight, Parts Count

All of these factors meet requirements and may improve through preliminary design (both configurations).

### 8.7 Control Forces

It has been shown in the foregoing analysis that the contractor has confidence that the specified goals can be met with both configurations, but it is believed that further effort should be spent in future developments to meet a more stringent goal. That is, a rotor such as the ITR should achieve control loads nearly identical to those of the system it is replacing in a retrofit application. For reasons cited in Section 7.11, this is necessary to avoid extensive redesign of the control system.

### 8.8 Mechanical Limiting Stops

At this stage of the design evolution, it appears quite clear that stops are not necessary on either configuration.

### 8.9 Maintainability And Operational Factors

These factors include installation, removal, inspection, blade folding, and environmental considerations such as heat, cold, ice, hail, humidity, salt, and lightning. Both EPB's should have superior characteristics for all factors mentioned. The sole exception is that the PEPB could prove more problematic in adapting to blade folding due to the unsupported torque tube. Extreme torsional loads are placed on the EPB during folding, and the CEPB torque tube snubber provides an ideal location for a pitch lock -- a simple hand-operated latch that must be activated by the ground crew before the folding pin can be removed.

### 8.10 Reliability

All factors leading to poor reliability have been avoided. One noteworthy example is the elimination of complicated, adjustment-hungry motion limiting stops. The pitch lock device introduced in Section 8.9 does not qualify as one of these since it is not an automatically operating, finely-tuned mechanism and in no way affects the operational qualities of the rotor.

## 8.11 Cost

Costs of both the PEPB and CEPB are considered low even though the loose criteria established for a goal were not quite met. It is expected that in final design, both concepts would be equal to or lower than other hingeless concepts.

## 8.12 Development Risk - Fabrication

Development risk in fabrication is considered low for both concepts due to the application of materials only in situations which are amenable to well-known fabrication techniques.

## 8.13 Development Risk - Test

Development risk in test for the CEPB is considered low because of the anticipated wide range in varying damping available for stability. Risk for the PEPB is somewhat greater because of the heavy dependence on elastic coupling for damping.

# 9.0 RECOMMENDATIONS

## 9.1 Concept Selection For Predesign

It is recommended that the Classic Elastic Pitch Beam be selected for development in the forthcoming preliminary design phase. The CEPB has all attributes necessary to a successful hingeless rotor head design and, in addition, presents a much lower development risk than the PEPB.

## 9.2 Emphasis In Preliminary Design

All of the factors used as criteria must be given the proper attention in the preliminary design phase; however, there are specific areas deserving of more emphasis These points are enumerated below.

9.2.1 Stability. It will be essential to characterize a variety of elastomers, both analytically and experimentally, for their contribution to damping requirements and for their molecular stability under harsh environmental conditions. An associated effort must be made to parametrically determine candidate mixes of elastic coupling and their role in damping for stability combined with the elastomer damping. Finally, candidate mixes of all parameters influencing damping that are proven by analysis to produce stable configurations must be parametrically evaluated with blade geometrical and structural parameters to determine impact on performance, vibration, and flying qualities.

9.2.2 Materials. There was evidence during concept definition that a hybrid of beam materials could produce benefits in lower cost and vulnerability for minimal weight and size increase. This possibility should be thoroughly explored through analysis, small specimen test, and full scale test.

9.2.3 Control Forces. The goal for reduced control forces should possibly be made more stringent considering the implications of eroding margins of safety of existing control systems.

9.2.4 Rotor Head Fairings. The CEPB more than meets established drag goals based on the simple calculations performed. It is recommended, however, that benefits in use of hub fairings (Reference 24) be traded off against cost and weight penalties in the preliminary design phase.

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# APPENDIX I

## ITR/FRR OBJECTIVES, TECHNICAL GOALS, AND SPECIFICATIONS

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March 1, 1982

## ITR/FRR OBJECTIVES

The overall objectives of the Army/NASA ITR/FRR Project are as follows:

a. To demonstrate a significant advance in rotor systems technology through the integration of the disciplines of rotor design, aerodynamics, structures and materials, dynamics and acoustics. The demonstration will show the potential for reduced life cycle costs; that reliability, maintainability, and survivability will be improved, that performance characteristics such as rotor L/D, fuel consumption, high speed maneuverability, agility and handling qualities are improved; and rotor weight, rotor noise and vibratory loads are reduced.

b. To demonstrate the improved technology to the extent that the major risks are removed, and to transfer this technology to industry for use in engineering development or product improvement programs.

c. To provide an advanced technology rotor, fully instrumented, having the capability for significant variation in rotor properties. This rotor will provide the capability to generate an expanded data base, and investigate further advanced in rotor technology.

## ITR TECHNICAL GOALS

One of the purposes of the ITR/FRR program is to stimulate the advance of rotor system technology to the maximum possible extent. While it is not appropriate to specify the degree of advancement as a requirement, reasonable technology goals can be defined to help stimulate and guide the technical thrust of the design. In what follows, where mention is made of vehicle system parameters or operating conditions, they are based on a design gross weight of 16,000 pounds, a vehicle flat plate drag area of 15.0 $ft^2$, and 4,000 foot pressure altitude, $95^oF$ conditions. Where technical goals are specified with respect to a baseline value, the baseline is taken to be the value corresponding to the UH-60 aircraft at 16,000 pounds gross weight.

| | |
|---|---|
| a. Maximum rotor equivalent lift-to-drag ratio, without hub drag, $L/D_e$, at $V_{Cruise}$. | 10.5 |
| b. Maximum rotor figures of merit, rotor alone. | 0.80 |
| c. Rotor hub flat plate drag area for a design gross weight of 16,000 pounds; for other values of the design gross weight, the goal for hub area is assumed to scale with the 2/3 power of the design gross weight. | 2.8 $ft^2$ |
| d. $V_{Cruise}$ using MCP of the powerplants required to meet the VROC performance capability specified in the System Design Specifications. For design gross weights different from 16,000 pounds the flat plate drag area is assumed to scale with the 2/3 power of the design gross weight. | 170 KTAS |

e. $V_{Dash}$ using IRP of the powerplants required to meet the VROC performance capability specified in the System Design Specifications. For design gross weights different from 16,000 pounds the flat plate drag area is assumed to scale with the 2/3 power of the design gross weight. — 185 KTAS

f. Reduction in low frequency impulsive noise from baseline (0-1000 Hz) when measured directly ahead of the helicopter in the plane of the rotor and when operating at an advancing tip Mach number of 0.9. — 6 dB

g. Rotor weight as a percentage of design gross weight. — 7.0

h. Rotor system parts count. — 75

i. Rotor system fatigue life. — 10,000 hours

j. Mean-time-between-removal (MTBR) — 1,500 hours

k. A vibration acceleration level based on a hypothetical estimate obtained by applying expected 4P hub vibratory forces and moments to a rigid body fuselage without anti-vibration devices. This vibration goal is defined within a volume extending to one-half the rotor radius in front and behind the center of the rotor, one-quarter of the rotor radius below the plane of the rotor, and one-quarter of the rotor radius laterally on either side of the rotor. The mass and inertia of the assumed rigid fuselage are to be taken equal to the baseline aircraft values or scaled appropriately if the design gross weight differs from 16,000 pounds. — 0.1g

l. The ITR rotor system will be designed to provide the lowest possible procurement cost for future production rotors based on ITR technology, without unduly compromising other cost factors that impact optimum life cycle costs.

ITR SYSTEM DESIGN SPECIFICATIONS

The following system design specifications are intended to establish a minimum set of operating conditions and other design constraints to be used to guide the design of the ITR.

a. Design Gross Weight

The ITR design gross weight shall be not less than 16,000 pounds and not more than 23,000 pounds. The specification requires that the ITR rotor be designed to have the thrust capability to permit the vehicle to hover OGE at 4,000 ft pressure altitude and 95°F with a total vehicle weight equal to the design gross weight plus a 10 percent fuselage download penalty.

b. Design Envelopes

For the purposes of the rotor design, the structural design envelope is +3.50g and -0.5g. The envelopes are shown in Figure 1. Slope landing conditions up to and including 12 degrees shall be accommodated.

c. Rotor System Instability

The rotor and test aircraft shall be free of critical aeroelastic instability mechanical instability at all operating conditions and throughout a typical range of gross weights. For the purpose of air/ground resonance instability, the rotor hub design requirements shall be consistent with fuselage and blade mass and inertia characteristics typical of the design gross weight.

d. Rotor Configuration

It is desired that the rotor be a four-bladed system. The hub design shall not preclude the incorporation of normal operational requirements for simple and quick manual blade folding and blade removal or replacement which does not require retracking or rebalancing. The blade design concept shall not be so restrictive or unconventional that it would be incompatible with the incorporation of provisions for meeting normal operational requirements including rain, ice, dust, and sand erosion, and lightning protection. Furthermore, the blade design concept shall not be incompatible with provisions for surviving limited tree strikes (one-inch pine branches), wire strikes (.25-inch copper non shielded wires), and combat damage (minimum probability of catastrophic failure following hit by 23mm HEI projectile).

e. Maneuverability

The aircraft shall provide the following capabilities at 4,000 feet pressure altitude, 95°F temperature, and at the design gross weight. From a level, unaccelerated flight condition at 170 KTAS, it shall be possible to attain, within 1.0 second from the initial control input, a sustained load factor of 1.75g in a symmetrical pullup. Following this load factor buildup, it shall be possible to maintain a minimum load factor of 1.75g for 3.0 seconds after the initial attainment of 1.75g. Airspeed at the end of the 1.75g, 3.0 seconds duration segment of the maneuver shall not be less than 140 KTAS. It shall be possible to attain, within 1.0 second from the initial control input, a sustained load factor of -0.25g in a pushover. Following the attainment of this load factor, it shall be possible to maintain a load factor of -0.25g for 2.0 seconds. At no time during either the pullup or pushover maneuvers described above shall angular deviations in roll and yaw greater than +10° from the initial unaccelerated level flight conditions be permitted.

f. Flight Test Aircraft

The ITR/FRR will be demonstrated in flight on a Contractor furnished aircraft, a Government bailed aircraft, or the RSRA. The Contractor will be free to propose the option of his choice. In the event the RSRA is not used for the demonstration testing, the Army may choose to carry out additional testing of the ITR on the RSRA. In any event, the Army and NASA intend to do research testing of the ITR and FRR on the RSRA.

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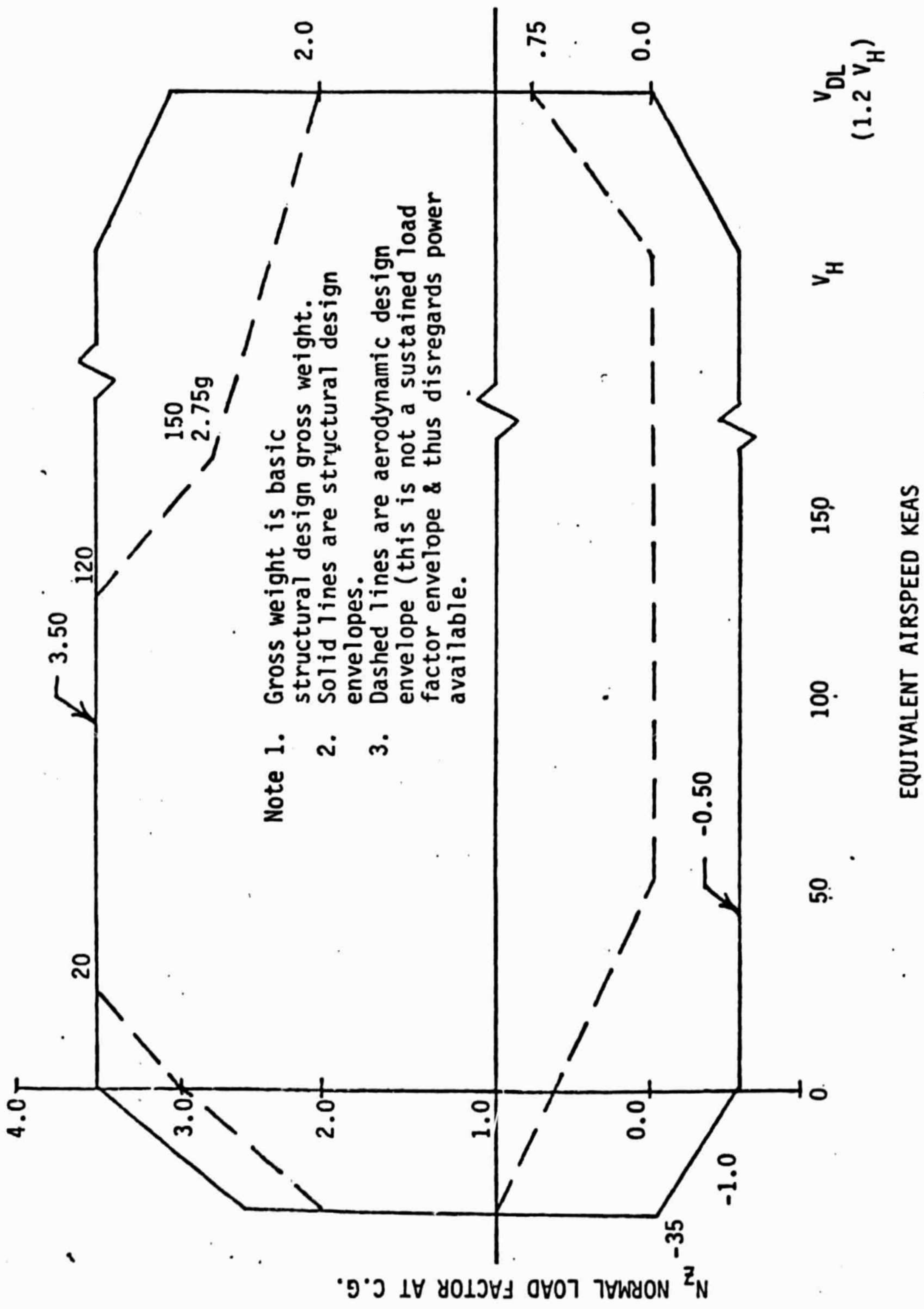


FIG 1. DESIGN ENVELOPES

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ROTOR HUB DESIGN SPECIFICATIONS

The following rotor hub design specifications establish minimum requirements to be used to guide the design of the rotor hub. The hub design specifications have been derived from the ITR System Design Specifications, specialized as appropriate for the development of hub components within the scope of the Concept Definition work.

Design Gross Weight - The ITR design gross weight shall be not less than 16,000 pounds and not more than 23,000 pounds. The specification requires that the ITR rotor be designed to have the thrust capability to permit the vehicle to hover OGE at 4,000 feet pressure altitude and 95°F with a total vehicle weight equal to the design gross weight plus a 10 percent fuselage download penalty.

Design Envelope - For the purposes of the rotor hub design, the structural design envelope is +3.50g and -0.5g. Slope landing conditions up to and including 12 degrees shall be accommodated.

Rotor System Instability - The rotor and test aircraft shall be free of critical aeroelastic instability mechanical instability at all operating conditions and throughout a typical range of gross weights. For the purpose of air/ground resonance instability, the rotor hub design requirements shall be consistent with fuselage and blade mass and inertia characteristics typical of the design gross weight.

Rotor Hub Configuration - It is desired that the rotor be a four-bladed system. The hub design shall not preclude the incorporation of normal operational requirements for simple and quick manual blade folding and blade removal or replacement which does not require retracking or rebalancing. The hub design concept shall not be so restrictive or unconventional that it would be incompatible with the incorporation of provisions for surviving limited wire strikes (.25-inch copper non shielded wires), and combat damage (minimum probability of catastrophic failure following hit by 23mm HEI projectiles).

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ROTOR HUB TECHNICAL GOALS

One of the purposes of the ITR/FRR Program is to stimulate the advance of rotor system technology to the maximum possible extent. While it is not intended to specify the degree of advancement as a requirement, reasonable technical goals can be defined to stimulate and guide the technical thrust of the Concept Definition work. Where the following properties are dependent on rotor vehicle system parameters they are based on a design gross weight of 16,000 pounds.

| | |
|---|---|
| a. Rotor hub flat plate drag area for a design gross weight of 16,000 pounds; for other values of the design gross weight, the goal for hub area is assumed to scale with the 2/3 power of the design gross weight. | 2.8 $ft^2$ |
| b. Rotor hub weight as a percentage of design gross weight. | 2.5 percent |
| c. Rotor hub system parts count, exclusive of standard fasteners. | 50 |
| d. Rotor hub moment stiffness. Defined by the moment in ft-lb, acting center of the hub, per unit angular rotation in radians of the rotor disc about an axis perpendicular to the rotor shaft axis. The rotor disc is defined by the circle inscribed by hypothetical rigid blade tips. The goal is specified for a design gross weight of 16,000 pounds; for other values of the design gross weight, the rotor hub moment stiffness goal is scaled in direct proportion to the design gross weight. | 100,000 ft-lb/radian |
| e. Minimum rotor hub moment. The minimum rotor hub moment in ft-lb, acting at the center of the rotor hub, below which fatigue damage will not be incurred by the hub; for a design gross weight of 16,000 pounds. For other values of the design gross weight, the minimum rotor hub moment goal is scaled in direct proportion to the design gross weight. | 10,000 ft-lb |
| f. Minimum rotor hub tilt angle. The minimum rotor disc angle defined in paragraph d above, below which fatigue damage will not be incurred by the rotor hub. | 5 degrees |
| g. Auxiliary lead-lag damping. The goal of the ITR is to develop a rotor system that does not require auxiliary hydraulic or elastomeric damper components incorporated in the hub. It is desirable to have the potential of incorporating some form of additional damping, if at some later stage in the development process it appears prudent to do so in order to solve an emerging stability problem. | --- |

| | |
|---|---|
| h. Torsional stiffness. The technical goal is to develop a rotor hub system that does not require substantially more blade pitch control actuator force than required by current rotor systems. | --- |
| i. Rotor hub system fatigue life | 10,000 hours |
| j. Reliability. Mean-time-between-removal (MTBR) for the hub. | 3000 hours |
| k. Manufacturing cost. The ITR rotor system will be designed to provide the lowest possible procurement cost for future production rotors based on ITR technology, without unduly compromising other cost factors that impact optimum life cycle costs. | --- |